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The ***SEPTEMBER, 1931***

SEP 2 1931

Heating · Piping and Air Conditioning

IN THIS ISSUE

Articles by

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EDGAR C. RACK

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JOHN C. REED

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EDGAR E. AMBROSIUS

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R. A. HANRIGHT

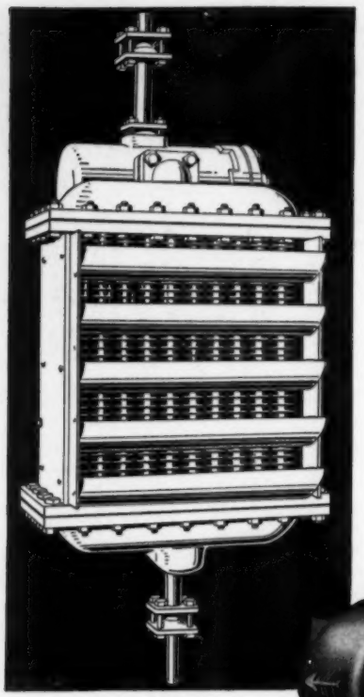
◆

Journal Section
of A. S. H. V. E.

AN ARTICLE ON THE INSTALLATION AND CARE OF GAGES OF INTEREST TO CONTRACTORS AND TO ENGINEERS.

HOW TO REDUCE AIR CONDITIONING COSTS BY IMPROVED COMPRESSOR OPERATION—A PRACTICAL PAPER.

Heating·Piping
and Air Conditioning



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The EDITOR'S PAGE

Heating · Piping and Air Conditioning

FROM ARTICLES IN THIS ISSUE

"Piping is arranged to supply the boiler feed pump through the deaerating heater from this condensate tank; thus, when the make-up plus the condensate exceeds the demand of the boiler feed pump, the piping is arranged so that the excess water may overflow from the condensate tank to a storage tank placed in the basement."—*R. A. Hanright.*

"Gages should receive the same care as a watch or clock. They should not be treated roughly or jarred. The case and glass should be cleaned and polished occasionally. A gage requires testing at regular intervals. After several years' use a gage should be thoroughly overhauled. This will assure longer life and a continuance of reliable service."—*John A. Masek.*

"The operating engineer may profit by some of these suggestions and thereby reduce the compressor losses, increase its efficiency and increase its capacity. This all results in a reduction in the cost per ton, an increase in tonnage and likewise an increase in net profits."—*John C. Reed and Edgar E. Ambrosius.*

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"For purposes of direct comparison between the results to be obtained by the use of the formulas, curves shown in Fig. 1 have been prepared. In arriving at what is termed the *correct thickness* of insulation for use, the annual saving per square foot of insulated surface is equated against the annual cost per square foot of applied insulation. The point at which these two curves intersect is then taken as the correct thickness of insulation for use."—*Edgar C. Rack.*

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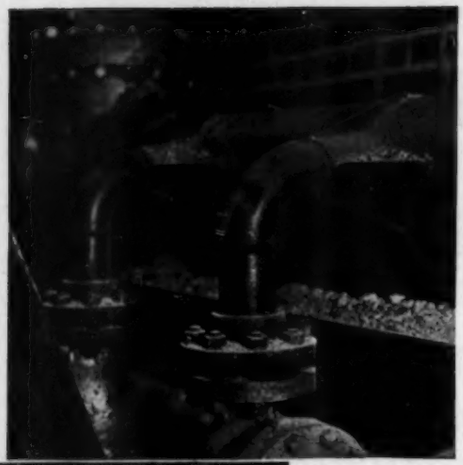

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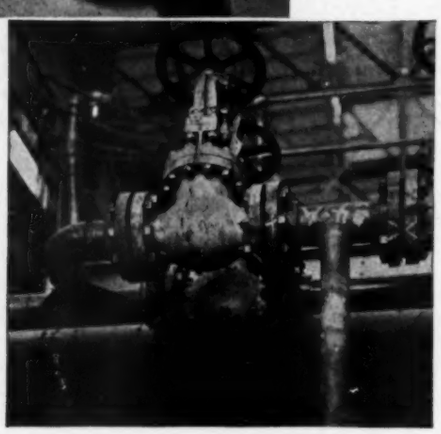

The Photographs:

Tube-Turns installed in the Bowling Green (Kentucky) plant of the Kentucky-Tennessee Light & Power Co., owned by the Associated Gas & Electric Company. W. S. Barstow Co., *Engineers.*

The welding was done in the fabricating shop and the field, *and the piping system is capable of withstanding 600 lbs. steam at 720° F.*

***Top:* Two 3", 90° Tube-Turns coming off main steam header. Van-Stone connections are for valves. *Middle:* 8", 180° Tube-Turns in water seal line. *Bottom:* 4", 90° Tube-Turn in boiler feed discharge water line.**

TUBE-TURNS

The Original Fittings for Pipe Welding

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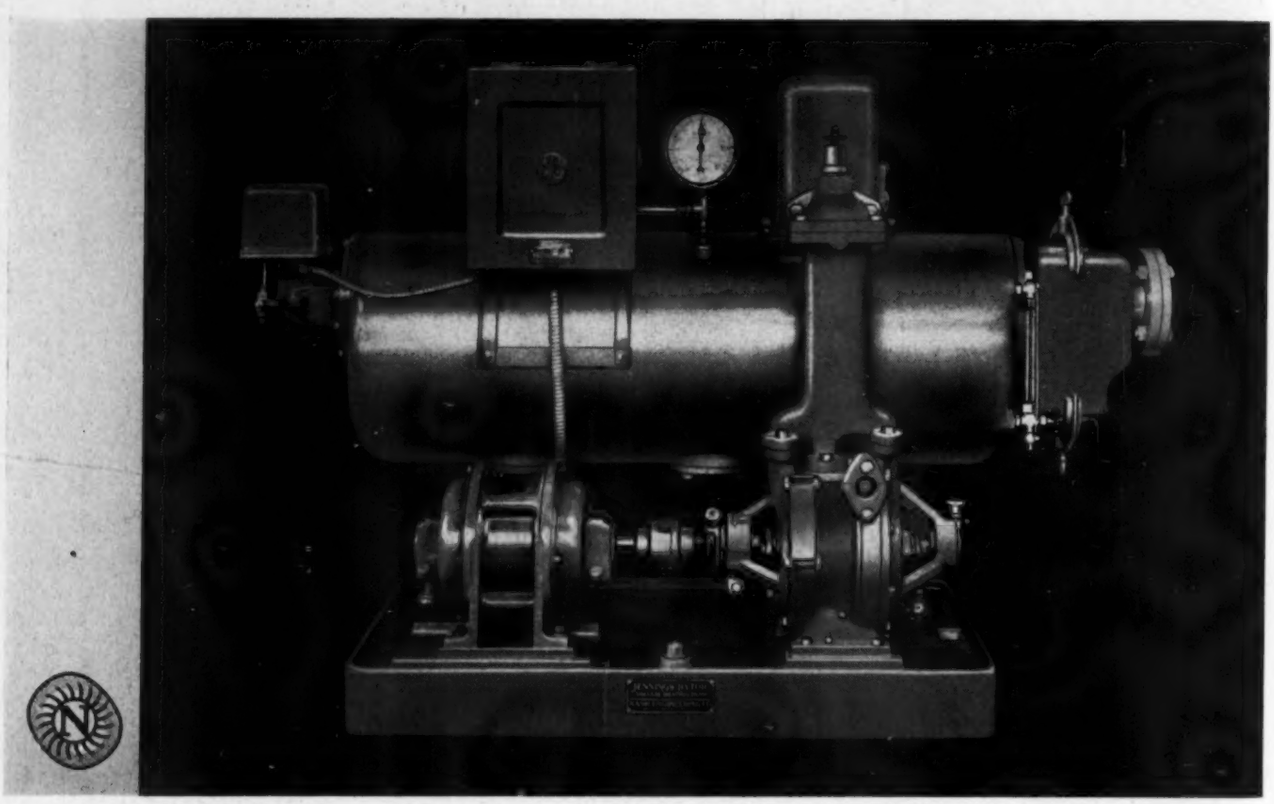

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Heating · Piping
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Vol. III | September, 1931 | No. 9

How to Install and Care for Gages

of the Bourdon Tube Type

By John A. Masek

Gages are costly and it is important that they indicate the conditions inside piping systems accurately at all times. For these reasons, it is essential that they be installed properly and that they be cared for carefully. The contractor and the operating engineer will find a wealth of valuable pointers in Mr. Masek's paper.

A PIPING system operating on a pressure or vacuum requires gages to denote conditions on the inside. The functions of gages are important from the standpoint of efficiency in performance and general safety; their uses warrant their installation wherever possible.

Gages of the bourdon single and double tube type are in general use. The combination of the proper tube, movement and dial graduations make up the following instruments; pressure, vacuum, compound, altitude, combination pressure and altitude, equalized altitude, thermometer, steam, duplex, ammonia, hydrogen, oxygen, hydrostatic and test gages.

Commercial Types of Gages

Flanged cased gages are made with iron or brass cases with rings of black finish, polished brass, nickel and chromium plated. The entire case and ring can be obtained in any of the above finishes.

Gages are identified by the diameter of the dials and are usually made in 3, 3½, 4½, 5, 6, 6¾, 8½, 10 and 12 in. dimensions. Dials with standard calibration are generally made up with a dull white finish and black markings and numerals. Other dial finishes may be in brass, dull silver and dull black with white figures. Small drawn steel or brass case gages are usually made in the following standard sizes: 2, 2½, 3, 3½ and 4½ in. diameters.

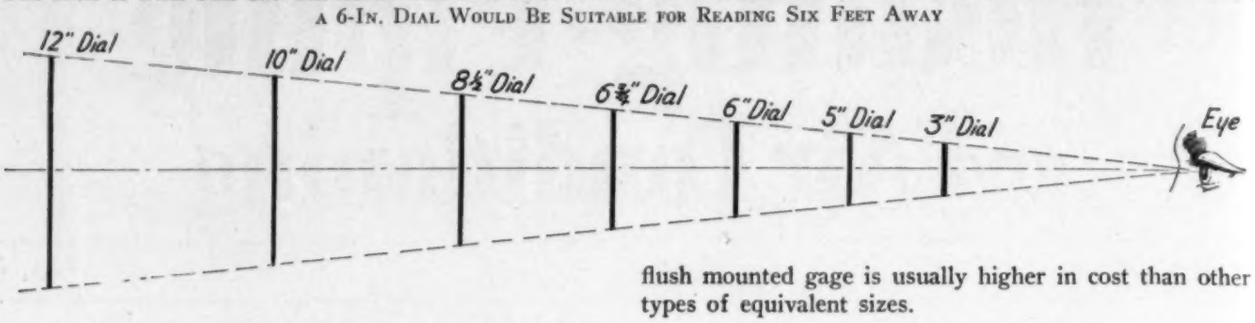
For gages graduated up to 1,000 lb., ¼-in. pipe thread is generally standard. Gages are furnished with bottom connection but where an instrument is to be mounted on a board or panel a back connection is provided. High pressure gages—such as used on hydraulic presses—which operate on pressures over 1,000 pounds are usually supplied with a ½-in. pipe thread.

Selecting the Proper Size of Gage

The size of gage to select depends upon the importance or value of the indication, cost and size of the machinery to which the gage is to be installed, etc. The larger the dial size the easier it is to read and greater accuracy can be obtained.

On small compressors or boilers a gage with a 3 to 4½-in. dial will usually be found satisfactory, while on

FIG. 1—RELATION BETWEEN THE SIZE OF GAGE AND THE DISTANCE AT WHICH IT MAY BE READ EASILY. FOR A GENERAL RULE, ONE INCH OF DIAL SIZE CAN REPRESENT ONE FOOT FROM THE POINT OF OBSERVATION FOR AVERAGE CONDITIONS. FOR INSTANCE, A 6-IN. DIAL WOULD BE SUITABLE FOR READING SIX FEET AWAY



larger apparatuses such as hydraulic presses, sprinkler systems, etc., the smallest dial size probably should not be less than 6-in. Large boilers, important pump lines, etc., should be equipped with not smaller than an 8½-in. dial and where it will be difficult to see, larger dial gages should be installed. Illuminated dial gages are suitable for night service or in dark engine or boiler rooms.

Fig. 1 shows how the size and distance affect the ease of reading in relation to the size of the gage.

Selecting the Style of Gage

The kind and type of case and ring selected depends upon the discretion of the engineer. Where the cost must be kept to the minimum iron case gages will often be found the lowest in price. The cost is greater for brass case and increases slightly with nickel and chromium plated finishes.

A surface mounted gage has a flange at the back of the case for mounting on a gage board, and in that case back connection should be specified; otherwise the gage may be made up standard with a bottom connection.

Where good appearance is desired of the instrument board, flush mounted gages leave only the front ring exposed while the case is at the rear of the panel. The flush mounted gage is usually higher in cost than other types of equivalent sizes.

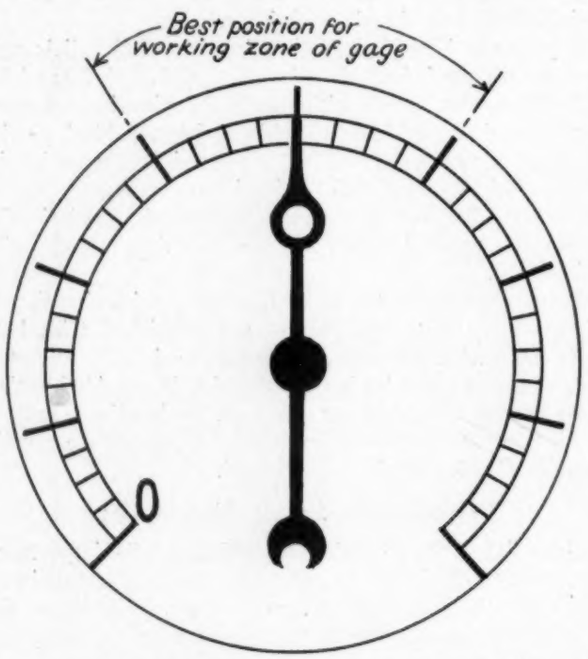


FIG. 2—A GAGE GRADUATED TO TWICE THE WORKING RANGE WILL USUALLY HOLD THE POINTER IN AN UP-AND-DOWN POSITION, WHICH IS ADVISABLE

Pressure Range of Gage

In order to get the best service from pressure gages it is very important to select an instrument with the proper dial calibration. A gage with a dial scale graduated to about twice the actual working pressure to which the gage will be subjected should be used. This will usually move the indicating pointer in an up and down or "twelve o'clock position" as shown in Fig. 2. Longer life with accurate service can then be expected.

The Single Spring Bourdon Tube Gage

A single spring bourdon tube pressure gage is shown in Figs. 3 and 4. The tube consist of a seamless drawn tubing—elliptical in cross-section as shown at *A-B*—and rolled in a half circle. The hollow socket is connected to the source of pressure at the pipe threads. The tip of the bourdon tube is closed and free to move.

Any pressure inside tends to straighten the tube, moving the tip toward the right, at *P*, transmitting the motion to the connecting link, then to the sector adjustment arm. The toothed sector turns on a spindle, transmitting the movement to the pinion, which is the axis for the indicating pointer. Change of pressure creates a motion which is multiplied and the indicating pointer moves over the scale of the dial which indicates the pressure in lb. per sq. in. at sea level. The hair spring prevents lost motion and keeps the joints of the movement pulled in one direction.

Vacuum and Compound Gages

The operation of a vacuum gage is the reverse of a pressure gage. When a vacuum is created the bourdon tube tends to collapse, which causes the end or tip to move toward the left at *V*. This motion is transmitted to the indicating pointer. The pointer moves in proportion to the vacuum present. The dial is generally graduated in inches of mercury.

Zero vacuum is the pressure of atmosphere at sea level which is equivalent to approximately 15 lb. absolute or 30 in. of mercury. Therefore, 2 in. of vacuum is considered, roughly, as one pound of pressure.

A compound gage registers both pressure and vacuum, or pressures above and below atmosphere. The pressure side is graduated in lb. per sq. in. while the vacuum side is graduated in inches of mercury. The construction of the gage is the same as that of the single spring pressure gage.

Most commercial types of gages are made with a bronze bourdon tube for medium pressures. Where pressures exceed 1,000 lb.—as on hydrostatic gages—a drilled

steel tube with threaded ends may be employed. Ammonia gages are usually furnished with steel tubes, as bronze tubing would be eaten through.

Double Spring Gages

Where pulsating high pressures are created in boilers and pump lines a double spring pressure gage will give steady readings as this type of gage cuts down the fluctuation of the indication pointer to a minimum. Fig. 5 shows the internal parts of the double spring with the bourdon tubes so arranged that the entire motion of the free ends is connected to the vertical bar at the tube tips. The horizontal connecting link transmits any movement of the tubes to the sector. The rest of the operation is the same as that of a single tube gage.

Effect of Freezing Temperatures on Gages

Where gages are exposed to freezing temperatures the double tube gage should be installed, as the bourdon tubes are self draining.

How Heat Affects Gages

A gage tube should not be exposed to temperatures exceeding 150 F. If this precaution is not taken as shown in Fig. 6, the high temperature in the steam and radiant heat will impair the resiliency of the tube and on high pressures melt the solder around the connections.

Gages on steam pressure should have a siphon installed between the gage cock and steam outlet. The siphon will become filled with condensation or water soon after the steam is on. This water will enter the tube, as shown in Fig. 7, and afford protection against steam. Where there is a certain amount of piping involved and a water leg is made as in Fig. 9 the siphon may be omitted.

Siphons either in the shape of a pigtail or U should be large enough to hold sufficient water to fill the bourdon tube when the gage is under pressure. It should be such that the water will not be withdrawn when the pressure is released, but leave enough at a level as shown in Fig. 8. Vacuum gages do not require a siphon but they should be installed with a compound gage when used on steam. Gages on corrosive gas lines can be protected by having a siphon filled with a light oil.

Gages on hot water services should be protected from excess heat. A water trap in the gage piping near the instrument will prevent the hot water from entering the bourdon tube.

Gages placed near extreme heat may become injured because of the oil in the movements becoming gummed, hindering the free action of the indicating pointers. When practical the gage should be attached some distance from boilers to a wall or column. The piping should be of ample size and the connection made direct from the steam chamber and not from the main.

All pipe connections leading to a steam gage should be steam tight, so that no steam will strike against the gage as it will work its way into the interior of the case and corrode the movement. Vacuum line piping connections must be tight to prevent the entry of air.

Gage Inlets and Cocks

A connecting or gage cock should always be placed between the gage and the pipe threads. This will permit the removal of the gage without the necessity of shutting down the plant. The cock can be used to throttle down the pressure leading to a gage if the indicating pointer vibrates considerably. Dampening the movement of the pointer will make it easier to read and will prolong the life of the gage. Cocks on vacuum gages should be wide open.

Pressure gages subjected to heavy and violent shocks as on car washers, hydraulic presses, feed pumps, and bilge pumps should be equipped with a restricted opening. Instruments with restricting screws are shown in Fig. 10. The gage with the large opening has the regular inlet. The restricting hole is drilled about 0.02 in. or smaller, depending on the system and service require-

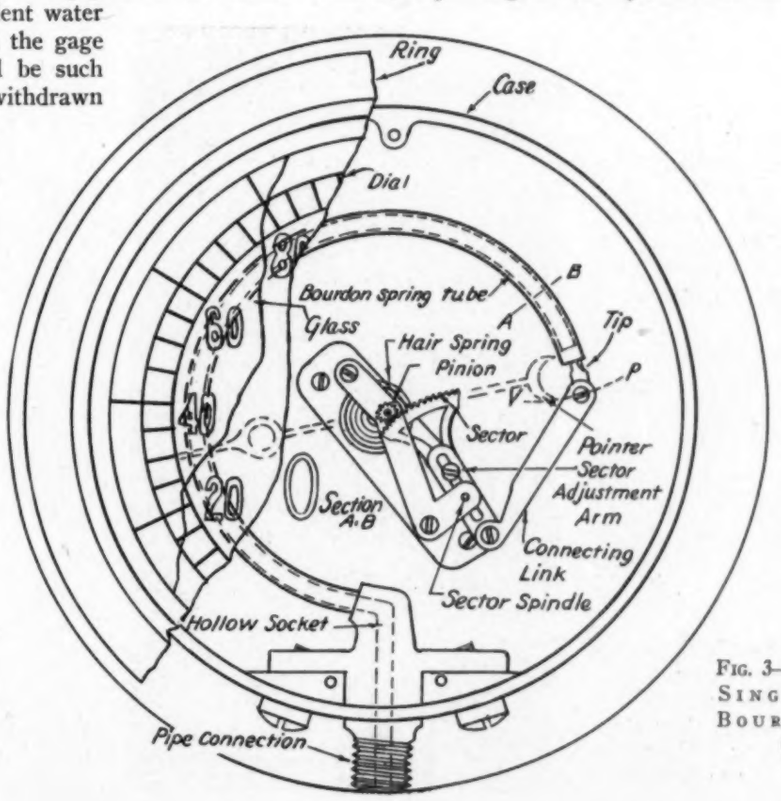


FIG. 3—A TYPICAL SINGLE-SPRING BOURDON TUBE GAGE

ments. This will remove shocks from the gage and help steady the movement of the pointer.

Gage Line Piping

A simple gage line connection consists of a pipe nipple as shown in Fig. 6 when the gage is to be placed over the outlet. This is used on hot water boiler, pump, and tank gages. On steam boilers a siphon is employed.

Gage line pipe thread openings should be made into screw fittings; when this is not provided for a hole must be drilled and tapped, either in the fitting itself or in the wall of the main pipe. It is good practice to make this opening one pipe size larger than the gage pipe line. Welding an extra heavy coupling to a main or fitting makes a good connection opening.

This connection should be made at the top of a main but not at the bottom, where there is a possibility of dirt or foreign matter getting to the gage, as in oil and process pipe lines. In such cases a strainer fitting should be installed. Protection is afforded in this way but care should be taken that the strainer is of ample size so it will not be the cause of giving a false gage reading. The strainer should be conveniently placed so it can be easily inspected and cleaned at regular intervals.

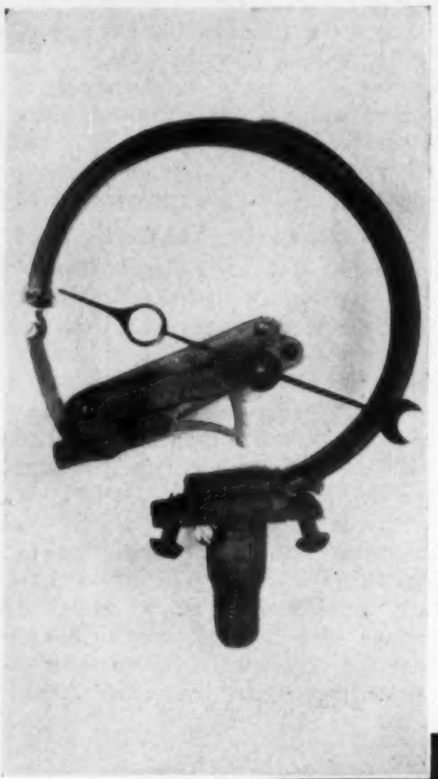


FIG. 4 (ABOVE) — TYPICAL SINGLE-SPRING BOURDON TUBE WITH MOVEMENT AND POINTER. THE CRACK WAS CAUSED BY OVER-PRESSURE

FIG. 5 (RIGHT)— DOUBLE-TUBE SPRING SHOWN INSIDE GAGE CASE WITH MOVEMENT AND POINTER

Gage Board Line Piping

When a gage is located some distance away from the source of pressure or vacuum (as for a gage board, which might be located in the engineer's office) the minimum size of pipe should not be less than ½-in. while ¾-in. pipe may be better. This will not have a tendency to restrict the gage pressure through corrosion or dirt settling in the pipe. The isometric drawing in Fig. 9 shows the method of piping for gages.

Such considerations as making provisions to permit blowing the gage line through with high pressures, a plugged tee for test gage connection and a drain cock near the gage should not be overlooked.

Correcting Gage for Water Column

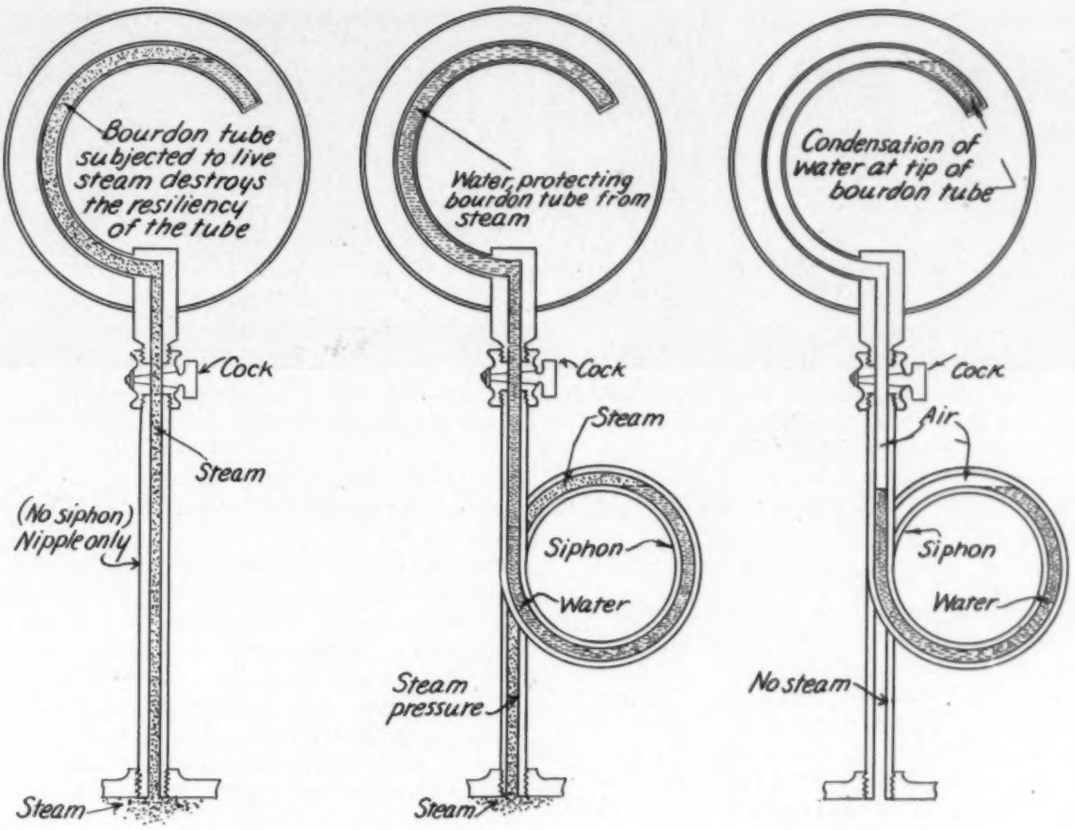
When steam gages are connected at the end of a vertical pipe (as shown in Fig. 9) which will fill with water of condensation, compensation must be made for the additional weight of the column of water. The pressure indicated on the gage is the actual pressure in the boiler plus the condensation in the pipe, which is equivalent to 0.433 lb. per sq. in. per ft.

A simple way to do this is to remove the ring and glass of the gage, take off the indicating pointer from the spindle with a hand jack as shown in Fig. 11, and replace so it points to zero. With some types of gages it is possible to adjust the pointer without taking it off by merely resetting a screw on the pointer. The gage so adjusted will then give the actual pressure carried in the boiler.

The Gage Board

A gage board is an assembly of several or many gages or instruments on a panel of slate, marble or other material in order to centralize gages in one place. A typical small gage board is shown in Fig. 12 with a schematic diagram showing the piping, valves and unions from the gage to the source.

The mounting of the panel can be accomplished as shown in Fig. 13 by using angle iron, drilled to accompany the mounting holes in the panel. Rosette head bolts are used to hold the panel in place.

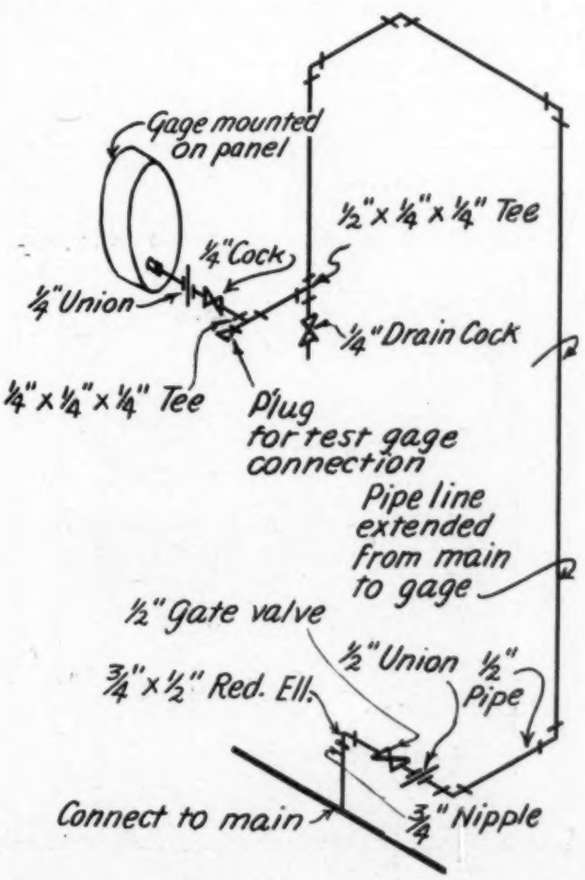


AT TOP

FIG. 6 (LEFT)—INCORRECT MANNER OF INSTALLING A STEAM GAGE

FIG. 7 (CENTER)—SHOWING HOW WATER IN THE SIPHON PROTECTS THE GAGE TUBE

FIG. 8 (RIGHT)—CORRECT MANNER OF PLACING THE SIPHON



AT LEFT

FIG. 9—PROPER PIPING FOR CONNECTING A GAGE WHERE THE GAGE IS SOME DISTANCE FROM THE OPENING

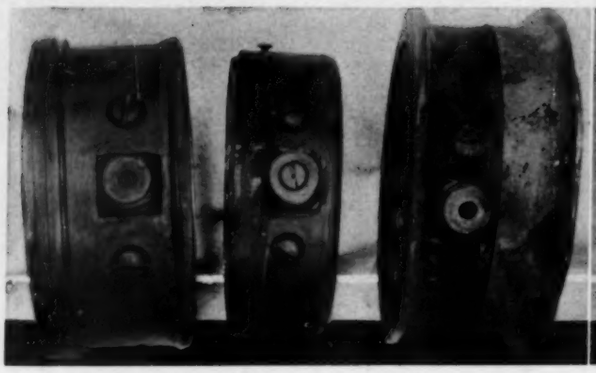


FIG. 10—THE TWO GAGES TO THE LEFT ARE TAPPED AND RESTRICTION SCREWS ARE SHOWN IN PLACE. THE GAGE ON THE RIGHT HAS A REGULAR OPENING

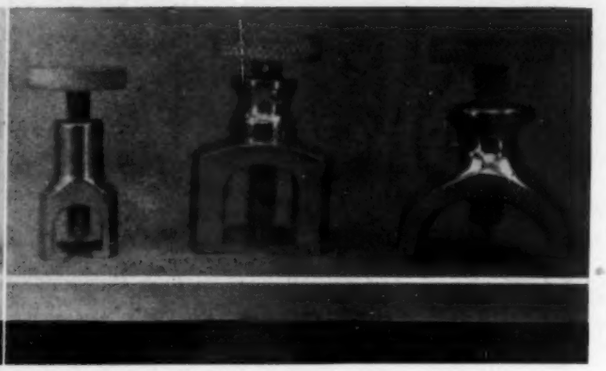


FIG. 11—HAND JACKS SHOULD BE USED TO REMOVE THE POINTER FROM THE SPINDLE PINION. THREE SIZES ARE SHOWN HERE

Small gage boards can be mounted on a solid wall by having special braces made.

Testing Gages

Gages should be tested every six months or oftener. They can easily be tested by comparison with a standard test gage. It is good practice to place a tee with a plugged outlet next to each gage. This simplifies the procedure for testing gages in service.

A gage giving incorrect readings should be sent to the manufacturer for adjustment or repairs. If there are a number of gages on an installation it pays to have a gage tester on the job.

Danger of Over Pressure

The normal pressure at which a gage should operate should be about one-half the pressure range indicated on the dial. Over pressure distorts and generally bursts the bourdon tube. Caution should be exercised that the proper gage graduation is used for each intended pressure.

Serious injury to human life as well as machinery has resulted from disregard of this point. In one instance, continually operating a hydraulic press on an over pressure caused the heavy steel bourdon tube to burst, which literally tore off a man's face. Careless manipulation of a high pressure steam reducing valve burst the tube of a gage on the low pressure side, blew out the glass and severely scalded a workman. Many cases can be cited where damages involving thousands of dollars have been caused by over pressures. Safety valves should always be on important pressure lines to prevent accidents.

Gage dials can be marked at danger points of operation, electric alarms attached to warn, or a maximum or tell-tale pointer utilized which will show the highest pressure reached. Thus, if the pressure is allowed to go too high, it is immediately detected.

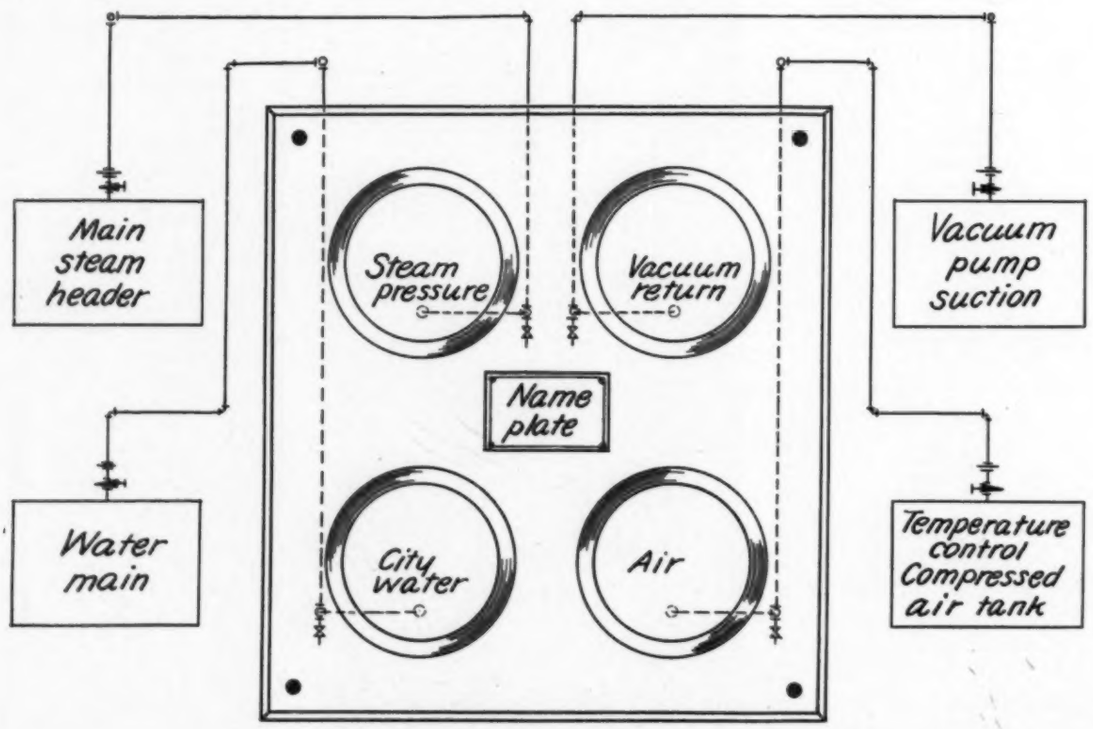


FIG. 12—TYPICAL GAGE BOARD ARRANGEMENT, SHOWING THE PIPING

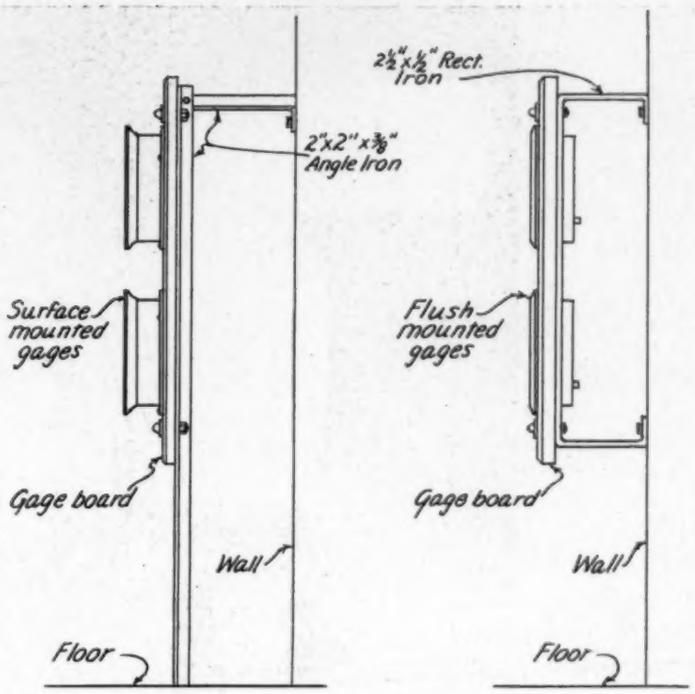


FIG. 13 — TWO METHODS OF MOUNTING GAGE BOARDS

Why a Gage Will Not Register

If a gage does not register there may be several causes. The opening at the pipe connection or the restricting screw may be plugged with dirt. In some cases the pipe line leading to the gage may have become filled with dirt, scale or some other foreign substance. The pinion spindle may be bent. Connecting screws sometimes become loosened or the movement may cause the parts to stick. Bourdon tubes sometimes become filled with whatever compound is in the system. Constant vibration pressures wear away moving parts and cause them to hold a certain position. Sudden or violent pressures may run the sector off the pinion.

Cause of Pointer Being Away From Zero

A new gage is sometimes received with the pointer away from zero. This can generally be attributed to rough handling in transit. The package containing the gage, if dropped, will cause the bourdon tube to set in the direction of the fall, which will alter the previous adjustment.

A steam gage without a siphon connected to it will cause the bourdon tube to lose its resiliency, destroying the gage.

The tube might be distorted by over pressure or a worn movement or hair spring being tangled might be responsible. High vacuum on a low pressure gage may collapse the bourdon tube. The tube might be lightened by corrosion.

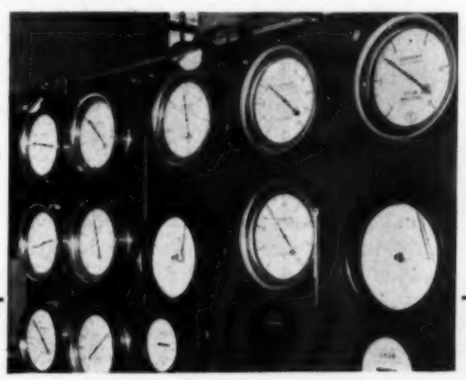
After a gage has been in use some time it may lose its resiliency and keep the pointer away from zero, which will cause the gage to show a greater pressure than actual.

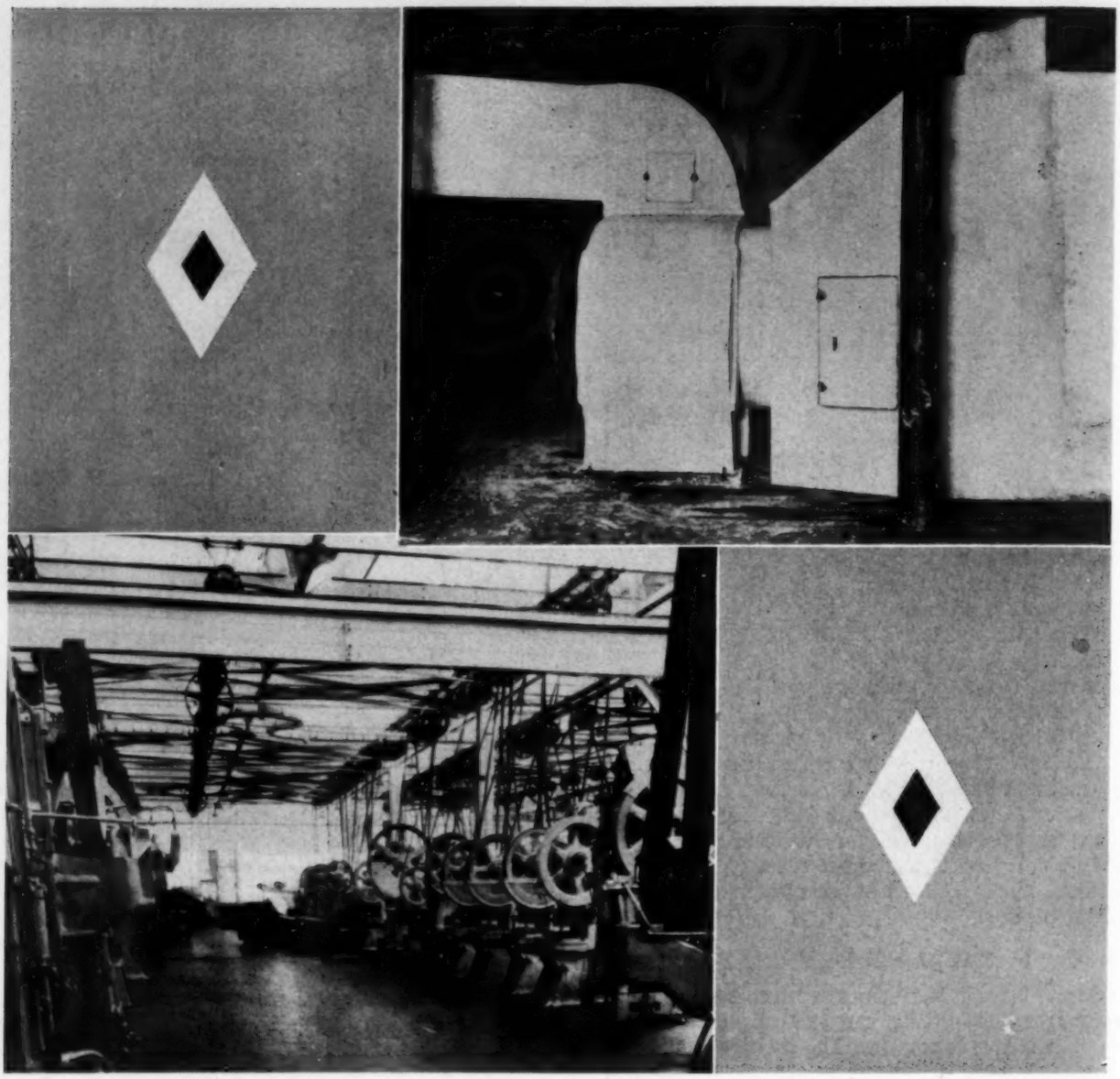
Leaks Inside a Gage

There are several possible causes of leaks inside a gage. Over pressure might burst the tube; a chemical or gas may eat through it. Leaks at the bottom piece are caused by solder melting away due to excess heat. Sometimes the cause is faulty soldering. A cracked tube may have been tempered too hard or the material may have been defective. Where a gage has been in use a number of years the tube may become brittle by constant motion and crack.

Gages should receive the same care as a watch or clock. They should not be treated roughly or jarred. The case and glass should be cleaned and polished occasionally. A gage requires testing at regular intervals. After several years use a gage should be thoroughly overhauled at the plant or sent to a reliable manufacturer. This will assure longer life and a continuance of reliable service.

If the glass becomes broken, it should be replaced as soon as possible in order to avoid injury to the pointer and dial, and to prevent corrosion of the movement and spring.





ABOVE—INSULATION OF AIR CONDITIONING EQUIPMENT IN A LARGE THEATER IN THE SOUTH. BESIDES CONSERVING HEAT, INSULATION IS ABLE TO AID IN MAINTAINING UNIFORMITY OF TEMPERATURE OF CONDITIONED AIR BEING DELIVERED TO THE AUDITORIUM AND OTHER SECTIONS OF THE BUILDING. BELOW—AN EXAMPLE OF THE CONTROL OF LIGHT BY MEANS OF WHITE PAINT. THE MACHINERY IS PAINTED WHITE ALSO. REFLECTION AIDS VISIBILITY AND THE UNIFORM DISTRIBUTION OF A SATISFACTORY INTENSITY OF LIGHT AIDS IN EFFICIENCY AND SAFETY OF OPERATION

Control of Heat, Light and Sound in Modern Buildings

By Edgar C. Rack*

ADVANCES in the art of building construction, as well as new developments leading to increased efficiency of apparatus and equipment, have made continued demands on the building material manufacturer for the bettering of materials. The industry has had also to keep abreast of progress by supplying newly developed materials as the evolution in building types and furnishings has progressed.

Broadly considered, the insulation sense of man includes those faculties which are susceptible to influence

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The use of insulation for controlling heat, light and sound in modern plants and buildings has developed rapidly within the past few years. The heating and air conditioning engineer is primarily interested, of course, in the first of these three functions, but it is felt that a discussion of insulation without referring to the reduction of sound and the control of light would be incomplete. Mr. Rack, therefore, discusses the broad subject of the utilization of insulation in modern buildings in an interesting and authoritative manner, of practical value to engineers.

by light, heat and sound waves. When an architect designs and lays out the fabric which encloses a building, he adjusts the materials of construction going into this fabric with due consideration for their artistic and economic values, and for the factors influencing human comfort, so that the completed structure will be a compartment with the fabric primarily acting as a medium to control light, sound and heat.

All forms of material required to aid or counteract the effect of these three forms of energy may offer complications so that before any material is determined upon for a specific purpose, it should be considered as regards one or more of its qualities which may influence its efficiency for other purposes. For example, a compartment constructed of a single layer of sheet metal would be satisfactory for privacy only insofar as it will effectively prevent the passage of light waves, and thereby insulate the occupants against visual disturbance or interruption of occupation.

Such a material considered as an insulation will serve as a very inefficient isolator of sound waves. It may even, under certain conditions, act as an amplifier of noise originating either within or without the compartment. This sheet of metal will act also as an extremely poor insulator against the flow of heat. From this example, it is seen that the insulation qualities of the material with respect to three forms of vibrant energy must be taken into consideration before definite specifications, outlining the construction details, are arrived at.

Concerns of the Designing Engineer

The engineer who designs the equipment for heating, ventilating, air conditioning and other apparatus and equipment for sundry purposes is generally concerned with the relative insulating value of the building's fabric when considered with respect to one or all of the above forms of energy. The illuminating engineer is primarily concerned only with those sections of the fabric which are capable of transmitting light waves. The heating and air conditioning engineer has usually to take into consideration the thermal resistance of all parts of the covering fabric of the structure. The acoustical engineer, who generally works in conjunction with the architect during the designing of the structure, is likewise concerned with the ability of all parts of the fabric, as well as the frame work, to resist the transmission of sound.

The reflection or radiation of these three forms of vibrant energy is also of more or less importance. The ability of the fabric of a building to radiate heat is probably of least importance, and is not generally taken into consideration, except in special cases of heating, ventilating or air conditioning equipment. It may be stated, however, that the relative radiating value of materials is becoming of increasing importance and research groups are devoting increased attention to learning more about such heat radiating factors. The reflective value of materials, insofar as light waves are concerned, is important and is most generally taken into consideration by the illuminating engineer in his problems of lighting design.

The reflective value of the material, with respect to sound waves, is also of prime importance. The acoustical engineer more often has to deal with reverberation of sound than with that sound conducted through the fabric or frame work of the structure.

It is intended here to treat primarily the subject of thermal insulation. However, the following brief reference to light and sound control should be of interest because of the interrelation existing between the quality of building materials to act as optical, acoustical, and thermal energy conducting or insulating mediums.

Light Insulation

Due to the fact that daylight is not uniformly constant in its intensity value over a twenty-four hour period, all buildings are equipped with artificial lighting to supplement and take the place of daylight. It is desirable that insulators against light be used for the purpose of obtaining necessary privacy. The utilization of these inefficient conductors of light further insures more efficient insulation against sound and heat.

There is one school definitely advocating windowless structures for the future. Logical basic ideas support

the suggestion and there is no doubt but that definite advantages are to be obtained. An economically designed wall construction offering a maximum resistance to the transmission of light waves, besides being capable of effective sound and heat insulation, can also furnish maximum structural strength, fireproofness and other advantageous features. However, at the present time the majority of us have definite and deep-rooted ideas of what our homes, clubs and offices should be like and while we desire improvements, we are reluctant to accept radical changes that are not in accordance with the ideals we desire. However, evolution will continue in building materials and structural methods. Future generations may see the architect and engineer designing and building windowless structures in which the greater part of work, rest and recreation will be carried out, or our future buildings may be practically all windows, as advocated by another "school."

A Windowless Factory

It may be of special interest to note that industry's first windowless building is, at the present time, being constructed in Massachusetts. This structure is being built at a cost in excess of a million and a half dollars, will cover five acres of floor space and will have solid walls and roofs unbroken by either windows or skylights. Elaborate equipment will be installed for the purposes of most efficient lighting, heating, ventilating, air conditioning and noise control. Special provisions are being made by painting, placement, etc., so that maximum light reflection will be obtained from the machinery and equipment in the building. By thus increasing the visibility of machines, the possibilities of accidents will be reduced. Special treatment will also be given the interior of the building fabric in order that the surface of this fabric may have favorable light reflective and sound absorbing qualities. Experiments conducted over a period of time indicate that there will be an increase in production-efficiency of as much as 35 per cent.

Sound Insulation

Sound insulation and control may properly be considered in two classes; one dealing with the transmission of sound and the other with reflected sound or reverberation and echoes. The engineer designing the equipment, and the architect working in conjunction with the acoustical engineer, can, with a high degree of efficiency, so plan the construction and arrangement of equipment as well as its placement, that its isolation from members and parts of the building will prevent the transmission of sound through the structural members and parts of the building to those places where it may act as a nuisance.

It has been more or less generally recognized in the past, that the elimination of both transmitted sound and reverberation is desirable for human comfort. It is now a proved and accepted fact that the elimination of unnecessary noise results in an economic saving due to increased human efficiency, comfort and health. While it can not be said that the art has reached such a state of perfection that sound transmission through building materials can be eliminated, at the same time, very decided progress has been made in the past ten years. It is now possible for the acoustical engineer to estimate with a close degree of accuracy the results to be obtained if his recommendations and specifications are followed in mechanical isolation methods and the proper placement of elevators, heating, ventilating and air conditioning equipment, and other possible sources of noise. The greater part of the noise now acting as a disturbing element to enclosed work or recreation centers, is that

A Good Example of the Treatment of a Ceiling for Sound Correction. Basements Can Be Converted Into Desirable Space for Occupancy by Accoustical Treatment of Walls and Ceiling, Proper Control of Light and Air Conditioning

noise originating within the compartment and such extraneous noise which may be transmitted through windows or walls from sources without the building.

Acoustical treatment of walls and ceilings is capable of reducing the noise level of sound originating in working compartments to a point where it can not be considered as a nuisance. The cost of application of such treatment can be definitely shown to pay for itself within a period of one to three years, due to increased human efficiency and the resulting higher economic producing power of the office or industrial worker. This result is capable of accomplishment by the use of treatments which are incombustible, vermin proof, permanent, capable of easy cleaning, and at the same time can be decorated so as to blend properly with the architectural treatment of the room. The light reflective value of the treatment, as well as its resistance to the flow of heat, can also be adjusted so that there will be no decrease in its sound absorptive qualities.

Heat Insulation

Advance in the art of manufacture can be measured for the past decade by the improvements made and developments brought forth. The building materials industry has accepted the demand made upon it and now, due to well directed research which has been and continues to be expended in the art, the engineer and architect can select materials at installed costs, varying from as low as four cents to as high as four times that price per square foot per inch of thickness.

Many engineers and others have asked why there are so many types and thicknesses of insulation from which to select. Some have gone so far in their explanations as to infer that the manufacturer is selling his product through the aid of vaguely defined insulation principles. Careful market research and the results of performance reports, however, indicate that the professional knowledge and experience of the engineer and architect qualify them to specify with intelligence. The owner also knows what is needed and is fully qualified to *buy* what he wants without being *sold*.

When Franklin invented the stove, buildings were, as a rule, built with thick walls which afforded a high degree of efficiency and at the same time were capable of storing considerable quantities of heat. Since that time and for economical and other obvious reasons, there has been a gradual decrease in the thickness of the building fabric. Since the pioneer research work of Peclet, made nearly one hundred years ago, investigations and tests of a varied nature have continued by physicists and engineers.

The results of these researches have enabled industry to raise the art of manufacturing acceptable materials above the status of hit-and-miss endeavor. The building materials manufacturer can supply efficient materials in various forms for meeting specified requirements and with the facts established in the laboratory, the engineer is capable of making computations of rates of heat transmission through compound walls, or other construction types, which are strictly more practical than theoretical.

Appraisal Factors for Heat Insulation Materials

The more important general requirements of building heat insulations grouped as nearly as possible in the order of their relative value are as follows:

1. *Conductivity and Adaptability*—Among the primary factors for consideration in the selection of a heat insulation is its ability to conserve fuel and promote human comfort by the establishment of uniform temperatures. This factor is mainly dependent on its high initial efficiency. Buildings are, however, constructed to furnish comfortable and safe occupancy for long periods of time. Therefore, if the material is not adaptable for the class of service imposed on it, due to decreased efficiency caused by settling, moisture absorption, or other failure, or on account of being otherwise unsuited for the service conditions to which it is subjected, it is evident that its high initial efficiency is of little value.

2. *Fire Resistance and Specific Heat*—It is becoming of increasing importance that building materials be classed as incombustible. With increasing congestion in residential, industrial and office locations, the building codes and local rating bureaus are requiring that insulation, as well as the other parts going into the fabric of structures, be highly fire resistant and non-flame propagating. Where a heating system operates continuously, the value of the specific heat of insulation is not of great importance. However, when the system operates at frequent intermittent periods, the value of the specific heat of the insulation in the building wall, as well as its placement, may be of primary importance. The time lag of heating systems and the effect of the placement of the components making up a building wall are being made the basis of intensive research studies at the present time.

3. *Moisture and Vermin Resistance*—There are obvious reasons why these factors should be considered as important. Moisture, besides reducing efficiency in the case of many organic materials, is likely to cause decay or promote bacterial growth. In the case of some inorganic as well as organic materials, where placed in contact with plumbing, electrical wiring, or other metal parts, a corrosive condition, with eventual failure of the metal, will result.

It is not necessary to comment on the sanitary as well as other reasons for the selection of materials resistant to mice and other vermin.

4. *Density, Strength and Workability*—The weight of an insulation may, to a greater or less degree, influence ease of installation, transportation costs, structural member strength and other factors. The factor of insulation strength should be considered where it may be used in exposed positions subject to knocks, and also where vibration may cause settling, opening of joints, or failure. When called upon to act partially or wholly as a support for finishes and equipment, or for other mechanical reasons, insulation should be more than casually evaluated as regards strength.

Workability influences the erection and installation costs. However, where alterations and modifications may be contemplated, ease of workability with a minimum of renewal costs on account of damaged or destroyed materials should also be considered.

It is realized that all of the requirements of an insulation insofar as their comparison to the perfect material is concerned would bring in many factors for discussion.

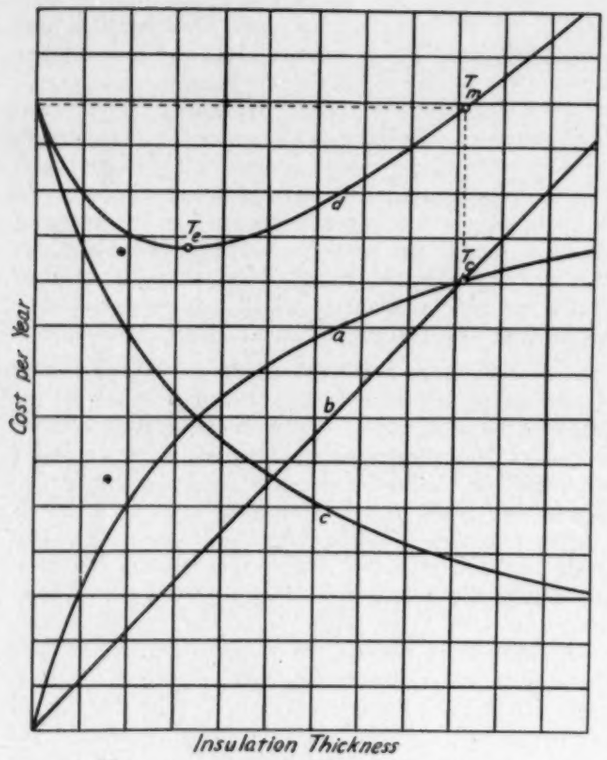


FIG. 1—ECONOMICAL THICKNESS OF INSULATION

However, the brief summary given is believed to cover the generalities for consideration in the majority of those structures which are not considered as special purpose cases.

Economic Considerations

The manifold values of heat insulation in all branches of industry are recognized and, on this account, there has been a continually increasing amount of work in the research laboratories for the past fifteen or more years, having for its aim the development of types and forms of insulation primarily suited for building purposes. It may be considered as a general assumption that the building of twenty-five or more years ago was sufficiently well insulated to act as an efficient heat insulator. But the value of heat has advanced, as has the demand for savings in the cost of heating. Due chiefly to economic considerations, insulations which are suited for industrial use at higher temperatures, and for periods of operation running 100 per cent to 400 per cent longer than most heating systems, can not be used for building insulation purposes.

Since the laboratory worker has established data on heat transfer which is generally recognized as reliable, more attention has been given to the economics of insulation and methods for determining the thickness to use in specific cases. It is not intended to go into a lengthy discussion of the variable factors bearing on the selection of the economical thickness. However, for convenience these factors are listed as follows:

Thermal value of fuel.
Overall efficiency of the heating system.
Total time of operation of heating system per year (duration of heating season).
Cost of fuel.
Thermal resistance of construction with which insulation is to be used.
Difference between inside and outside air temperatures during heating season.
Conductivity of insulation.
Installed cost of insulation per inch of thickness.
Annual return desired on installed insulation cost.

Contributors on the economics of insulation have included all the above factors in their presentations on the subject. The majority of the methods, however, which have been presented, are of a graphic nature for purposes of convenient determination. For the past eight years estimators have generally used the rational analytical method derived by the late L. B. McMillan,[1] which determines the *economical thickness* of insulation for use with any given conditions. P. D. Close[2] has recently presented a formula which is offered for determining the *correct thickness* of insulation. It should be noted that in the use of the *correct thickness* formula it is necessary in all computations to assume as constant: (1) the average calorific value of coal, (2) the over-all efficiency of the heating system, and (3) the duration of the annual period of operation of the heating system. The latter contributor has more recently[3] presented his formula in a revised form. A numerical constant which includes these three factors is included in the formula and is given as 4.2 in the second and 3.23 in the third reference cited. No explanation is offered for this apparent revision of the formula; however, there are reasons for believing that the first and second factors listed above have been increased in assumed average values by 8.33 per cent and 20 per cent respectively over the values originally selected to represent average conditions.

In making calculations it is generally felt that all factors which vary in their limits, as will the above, should be left to the judgment and knowledge of the estimator for his evaluation and use. It is also of particular note that the *correct thickness* formula is limited in its application for use only when coal is used as a fuel, while McMillan's *economical thickness* formula is applicable for any class of fuel.

Economical Thickness of Insulation

For purposes of direct comparison between the results to be obtained by the use of the McMillan and Close formulas, curves shown in Fig. 1 have been prepared. In arriving at what is termed the *correct thickness* of insulation for use, the annual saving per square foot of insulated surface *a* is equated against the annual cost per square foot of applied insulation *b*. The point T_e at which these two curves intersect is then taken as the *correct thickness* of insulation for use.

McMillan adds the annual cost of heat loss per square foot *c*, which decreases with the addition of insulation, and the annual cost per square foot of applied insulation *b*. The thickness at which the sum of these two costs *d* is a minimum is obviously the *most economical thickness* T_e.

It is of interest to note that the thickness T_e where the

[1]The State of Existing Data on Heat Transfer Through Insulation, Vol. 48, Trans. *A.S.M.E.*, 1926.
[2]How Much Insulation to Use, *The American Architect*, March, 1930.
[3]Economics of Thermal Building Insulation, Part I, *Heating and Ventilating*, August, 1930.

savings curve *a* intersects the applied insulation costs curve *b*, coincides with the thickness T_m on the total cost curve *d*. It easily can be proved that T_m is the thickness of insulation for which the total annual cost per square foot is equal to the annual cost per square foot of heat lost through the construction when no insulation is used. Obviously, if it is desired to apply sufficient insulation so that the total cost per year will equal the cost per year through the construction without insulation, the so-termed *correct thickness* formula offers one convenient method for arriving at the value of that thickness.

Comparison of Correct and Economical Thickness

Fig. 2 has been prepared in order to visualize the difference in thickness arrived at by the use of the two formulas. All factors have been assumed as equal for use in both methods of calculation, and the resulting graphs show the increased thickness of insulation required for increasing coal costs. It is seen that the two formulas give identical results only at that point where the cost of coal is sufficiently low in cost to preclude the use of insulation. It is further seen that the formula for determining the so-termed *correct thickness*, T_c, gives results which are increasingly divergent from those obtained by using the *economical thickness*, T_e.

It is not considered necessary to repeat the formulas nor their derivations in this brief discussion, as detailed explanations are given in the references cited. However, in repetition, it must be emphasized that an exact comparison between the *correct thickness* formula (using 3.23 as constant), and the *economical thickness* formula, will apply only when the following conditions and concrete numerical values are assumed as the same for both methods of thickness determination:

5,040 hr.=annual period of duration of heating season.
60 per cent=overall efficiency of heating system.
13,000 Btu=thermal value per pound of fuel.
Coal *only* to be used as fuel.

If one or all of these numerical values are changed, the characteristics of curve T_e may be materially altered, but curve T_c remains unchanged. Obviously, the *correct thickness* formula is inapplicable when any other fuel than coal is the source of heat.

A still more recent consideration of the economics of insulation by Close[4] has included a formula for estimating the thickness of insulation, where the basis for determination is the desired instantaneous return on the investment. This latest formula (for coal heated buildings only) as presented, is equivalent to McMillan's *economic thickness* formula, with the exception that the three variable factors heretofore referred to are continued to be considered as constant and are grouped in the single numerical value (3.23) which is included in the *correct thickness* formula.

As stated, many contributions have been made on the economical thickness of insulations, and for this reason it is not felt necessary to go into a more detailed discussion of the subject now. It is believed, however, that even among engineers there is much confusion existing with reference to the principles involved, and the manner of making calculations for the economic thickness of insulation. In determining the economical thickness of insulation, the principles involved are

[4]Economics of Thermal Building Insulation, Part II, *Heating and Ventilating*, October, 1930.

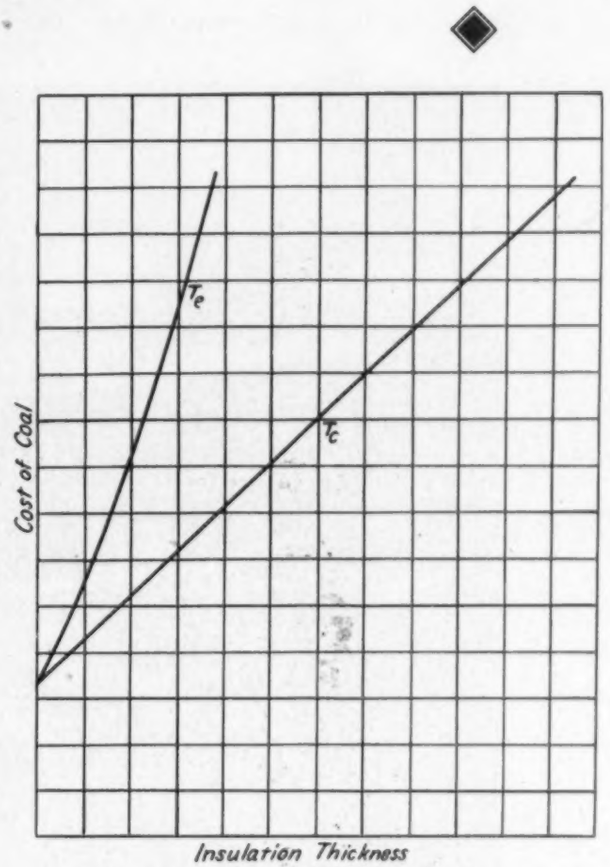


FIG. 2—COMPARISON OF SO-TERMED CORRECT AND ECONOMICAL THICKNESS OF INSULATION

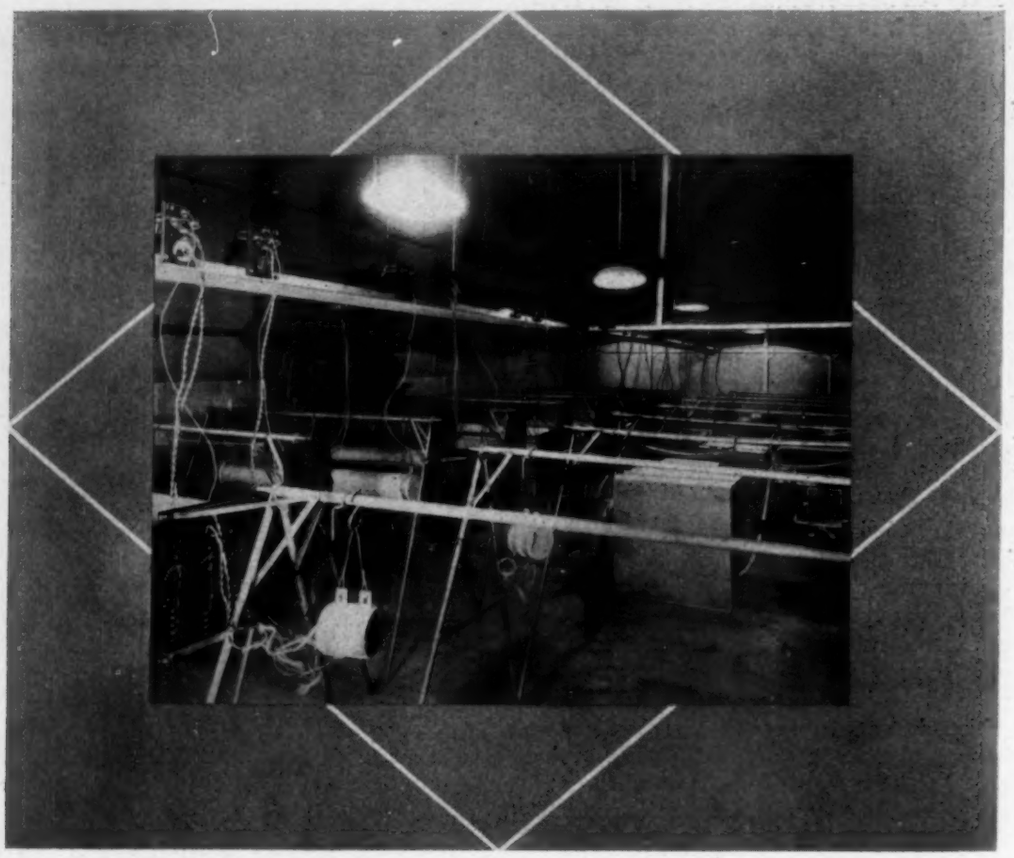
exactly the same as those involved in the determination of the economical size of a piece of equipment. In making such determination, the result desired is the maximum net monetary saving for any given condition. *The correct thickness of insulation to use in any case, should be that thickness for which the last increment added will pay a required minimum return on the additional cost of that increment.*

Research and Materials for the Future

The physicist in his laboratory is, at the present time, intensively continuing his work on the study and effect of surface temperatures. This is a factor which has been brought to the front with increasing emphasis in recent years. The engineer has, in the past, primarily stressed the maintenance of satisfactory temperatures at the breathing zone, but it is now recognized that the efficiency of the fabric of a building is a measure of the effective maintenance of human comfort. The industrial research and development organizations are also studying this factor in order to apply it to advantage in the form of specifications as to thickness and placement of building materials, and also with reference to the specific heat of the materials, as well as their surface finish, with the main idea in mind of improvements and developments.

The scientist is likewise making solar radiation the subject of study. The results of these investigations are being studied by industry, as well as by the engineer, with much interest. Calculations involving power re-

INDUSTRIAL RESEARCH LABORATORY DEVOTED EXCLUSIVELY TO PROBLEMS DEALING WITH HEAT INSULATING MATERIALS



quirements in many heating and air conditioning problems necessitates taking the solar radiation factor into consideration, and industry can well look to the physicist's work with the object in view of possible modifications, alterations, or development of improved materials and specifications for their use.

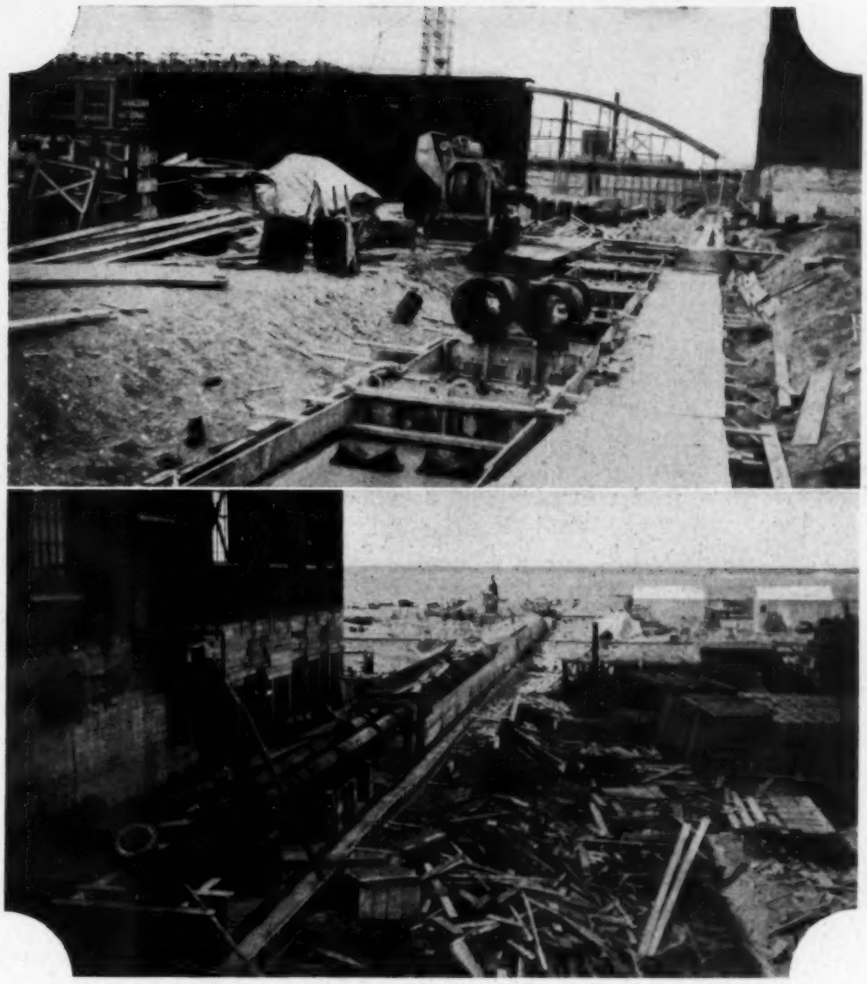
With respect to wall transmission units, studies being made by research organizations working independently, as well as those cooperating with industry, are being continued. Further standard units will be established for use in particular localities, with due regard to weather conditions and local material specifications.

A decrease in the thickness and weight of the covering fabrics of structures would mean less steel or other structural material essential for support. A greater advantage would probably be the increase in floor area and the resulting higher net return to the owner. Intensive study and research tending to the development of a new type of structure is being carried on, and industry is heartily cooperating in order to be able to supply promptly, if and when wanted, those materials which can be erected with the least cost.

There is also an apparent tendency toward a type of construction utilizing metal as the fabric of the future building. Metal in the outer skin of building fabrics has been applied to some extent in two of the largest and most modern of structures, the Chrysler and the Empire State office buildings in New York City. Although metal walls and floors, as at present proposed, would not provide all the factors necessary for human comfort desired with the ideal structure, it is fairly safe to say that the building materials industry can offer specifications and materials at the present time which will provide lighting efficiency, sound control, and thermal insulation in a degree superior to the ability which it is believed will be required by the designer in the utilization of all metal for the skin of the building fabric.

Research in Building Materials

Highly efficient materials, which may be strictly termed modern, are available and industry can supply them, or modify them to meet such methods and processes which the designer may wish to adopt in evolving the modern type of structure. However, industry is not waiting, but is alert and working on improvements. The interior of one section of an industrial research laboratory, which is devoted exclusively to problems dealing with heat insulating materials, is pictured above. This laboratory is one of the largest, and is only one of many in the United States capable of working at conditions of inner or outer insulation surface temperatures varying from temperatures well below those used in commercial refrigeration work to those well up in the range of temperatures at which modern boiler refractory materials are operated.



ABOVE—TUNNEL FLOOR POURED WITH EXPANSION JOINTS SET IN PLACE ON CONCRETE PADS. BELOW—PIPES LAID IN TUNNEL AND SIDEWALLS AND ROOF PARTLY COMPLETED

Piping Reliability—

of Importance in Power Plant

Near Arctic Circle

By R. A. Hanright*

ON THE west coast of the Hudson's Bay, almost within the fringe of the Arctic Circle, the first unit of an eight million bushel grain elevator is being constructed for the Department of Railways and Canals at Fort Churchill, scheduled to be ready for operation Sept. 1, 1931. A power plant will supply the electrical energy to operate the motors driving the elevator machinery, and the steam for the dryers and heating. The elevator and power house are situated approximately 600 ft. from the harbor front, with a separate pump house at the harbor front, as shown in Fig. 1.

Steam Superheated to 650 F

The power plant contains three boilers with integral economizers designed for a working pressure of 275 lb. Two of these units are 600 h.p. each, and the other 300 h.p. Each boiler is fired by a chain grate stoker,

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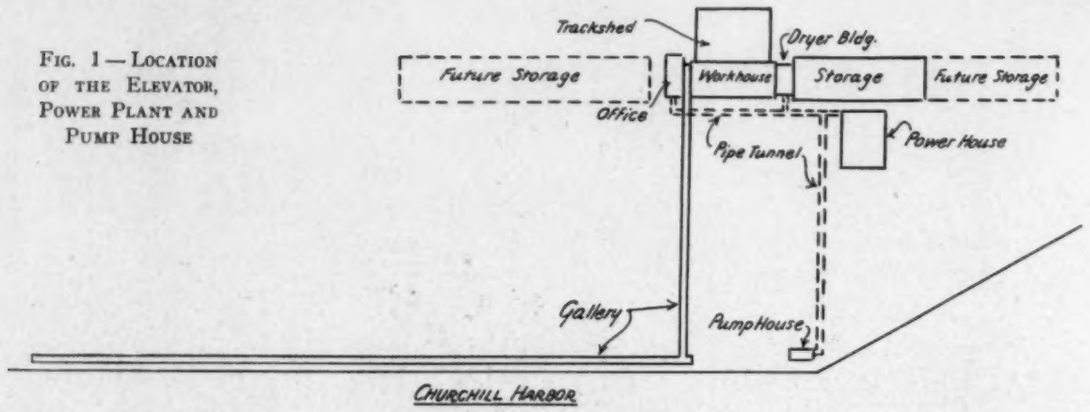

FIG. 1 — LOCATION OF THE ELEVATOR, POWER PLANT AND PUMP HOUSE

with furnace designed to burn western Canada coal. Each is equipped with a superheater to give a final steam temperature of 650 F. The arrangement of the boilers, stokers, fans, coal and ash handling equipment, and a portion of the piping is shown in Fig. 2.

Two 1,500 and one 500 kw. turbo-generators, designed to operate condensing, are installed. An evaporator is included for make-up feed water. In conjunction with the closed heater, deaerating heater, condensate and storage tanks it operates on an overflow principle.

The boiler feed pumps are capable of pumping 300 g.p.m. at 215 F, one being motor-driven and one steam turbine-driven. In the pump house are three pumps for delivering cooling water to the condensers, two of which are motor-driven and one gasoline engine-driven for standby and initial operation. Fig. 3 is a plan view of the pump house. This view also shows the arrangement of piping in the pump house and the arrangement of the water intake and discharge wells, with the position of the sluice gates indicated.

Concrete Pipe Tunnel

A concrete pipe tunnel extends from the power house to the elevator building, with a branch tunnel to the pump house. They are especially designed, being built on piles driven on approximately 9-ft. centers under each side

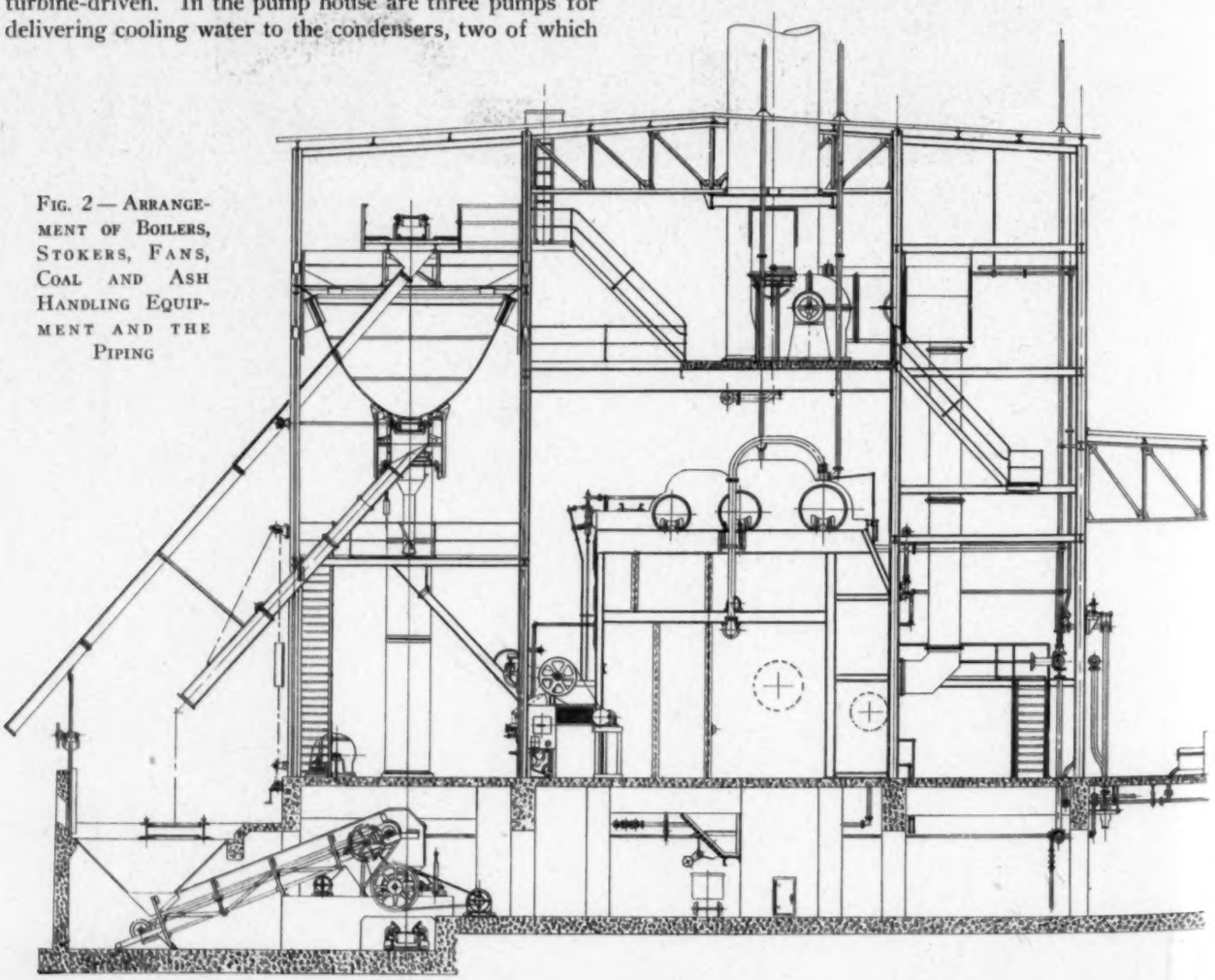

FIG. 2 — ARRANGEMENT OF BOILERS, STOKERS, FANS, COAL AND ASH HANDLING EQUIPMENT AND THE PIPING

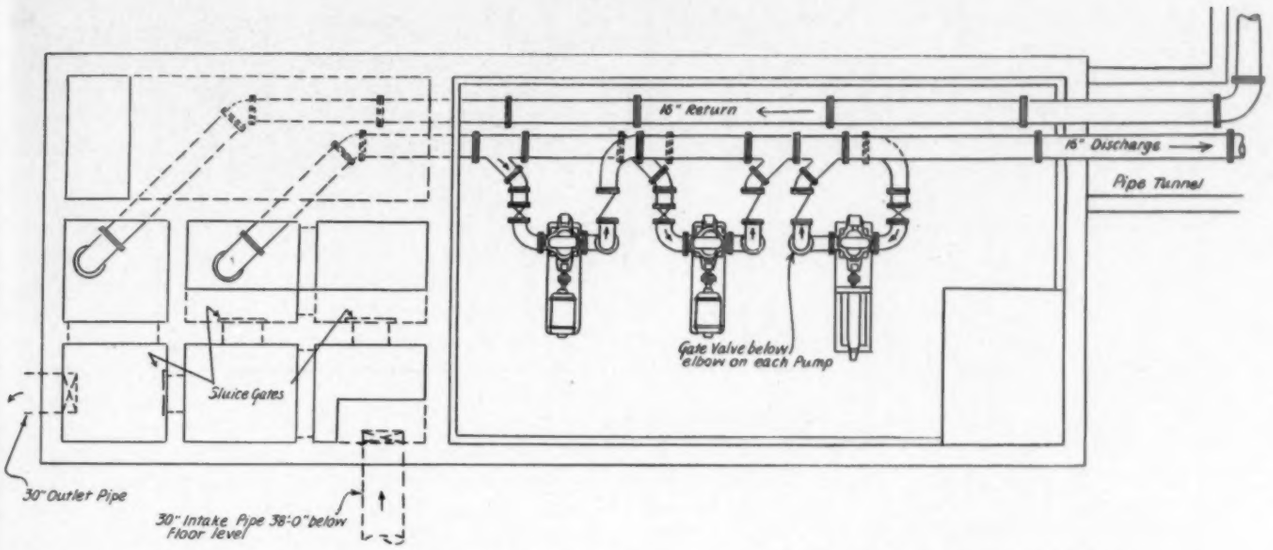

FIG. 3—PLAN VIEW OF PUMP HOUSE, SHOWING PIPING AND WATER INTAKE AND DISCHARGE WELLS

wall, with expansion joints for each 100 ft. of length; each joint is made with flashing, the construction being indicated in Fig. 4.

Two 16-in. cast iron pipes are located in the tunnel running from the pump house to the power house; one furnishes cooling water to the turbine condensers, raw water to the evaporator and water for general service. The other is the return line. They are set in cast iron saddles spaced on 12-ft. centers, and slope toward the pump house so that they may be drained dry. The cast iron saddles are set on concrete piers built up from the floor of the tunnel to proper height to accommodate the slope. There is a 600-ft. length of straight pipe in the tunnel; both pipes are anchored approximately 30 ft. from each end and a 16-in. double expansion joint designed for sea water is installed in the center of each run.

These cold water pipes are covered with a layer of asphalt paper enclosing the two pipes, over which is placed three layers of hair felt—each 1-in. thick—with asphalt paper between each layer. The outer surface is covered with two layers of asphalt paper, held in place with marline twine, and the outer covering has a heavy coating of asphalt paint. Between the two pipes and inside the covering is installed a bare steam pipe, running the full length of the cooling water pipes in the tunnel and to the pump house, where it furnishes steam for heating. Fig. 5 is a section of this arrangement. Fig. 6 is a complete diagram of the piping for the power house, exclusive of the piping for heating and compressed air.

By-pass from Main Steam Header to 60-Lb. Line

It will be noted from Fig. 6 that the piping from the superheater is connected to a 14-in. O. D. header for distribution to the turbines and other apparatus. A desuperheater is used for apparatus where saturated steam is required and reducing valves are employed for steam to auxiliaries, and also for a by-pass connection from the main steam header after the desuperheater to the 60-lb. steam line which originates at a point where the steam is extracted from one of the main turbines. This by-pass will be used when the turbo-generator is off the line or when the amount of steam being extracted does not supply the demand for evaporating and heating.

Make-up Feed Water

Fig. 6 also shows the piping necessary in conjunction with the evaporator when using salt water to manufacture make-up boiler feed water. Raw sea water is piped through closed heaters to the evaporator, the condensate being trapped and sent under steam pressure to the deaerating heater. The vapor from the condenser is piped through the closed heater, where in heating the raw sea water before entering the evaporator approximately one-quarter of the vapor is condensed; a lifting trap is employed to send this condensate to the deaerating heater. The remainder of the vapor goes to the 12-in. exhaust header and into the deaerating heater.

FIG. 5—A BARE STEAM PIPE RUNS BETWEEN THE WATER PIPES AND FURNISHES STEAM FOR HEATING

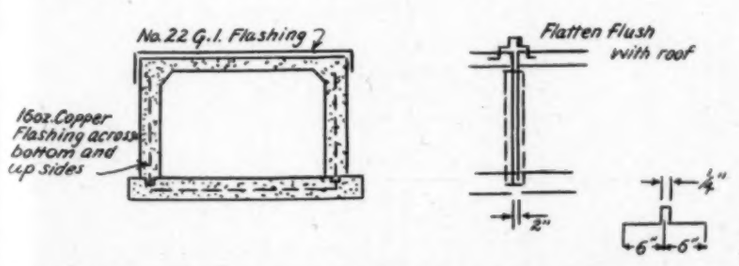

FIG. 4—CONSTRUCTION OF EXPANSION JOINTS IN CONCRETE PIPE TUNNEL

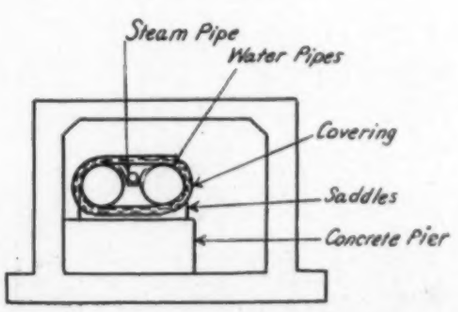

Table 1—Piping Schedule

Service	Pipe	Fittings	Flanges	Valves	Type	Gaskets	Unions—Screwed	Bolting	Covering	Painting
250 Lb. Steam, 650 F Total Temp.	Seamless—Open Hearth 14″ Header—.5″ Thick Wall 8″—.344″ Thick Wall 6″—.313″ Thick Wall 4″ and under—Std. Wt. Steel	Ex. Hy.—C.S.—Flanged	3″ and over—Van Stone Type C. S. Flanges—F. & D. to Amer. Std. for 250 Lb. 2½″ and under—C. S.—Screwed and Weld.—F.&D. Amer. Std. for 250 Lb.	2″ and under Ex. Hy.—C. S. Body—Monel Metal Trim—Rising Stem 2½″ and over Ex. Hy. C. S. Flanged O. S. & Y.—Monel Metal Trim 6″ and over with By-Pass	Gate—Flg'd Globe—Screwed Gate Gate	1/16″ Thick	None	Studs and Stud Bolts—Alloy Steel—Medium Carbon Steel Nuts—Hex.	1¼″ Inner Layer—High Temp. Sect. Covering—1½″ Outer Layer 85% Magnesia Sect. Covering 8-oz. Canvas Jacket—Painted	One Coat Red Lead and Oil
60 and 125 Lb. Steam not exceeding 450 F	12″ and under, Std. Full Weight Steel	3½″ and under, C. I. Screwed 4″ and over C. I. Flanged	C. I. Screwed—F. & D. to Amer. Std. for 125 Lb.	3½″ and under I. S.—Rising Stem Bronze—Screwed 4″ and over—O.S.& Y. C. I. Body Flange—Comp. Mounted	Globe Gate Globe Gate	1/16″ Thick	2″ and under—M. I. Brass Seats (Use Flanges over 2″)	Std. Steel Bolts—Sq. Hd.—Hex. Nuts	1½″ Thick Air Cell Sectional Covering—8-oz. Canvas Jacket—Painted	One Coat Red Lead and Oil
Vacuum and 0-25 Lb. Steam	12″ and under Std. Wt. Steel 14″-20″-⅜″ Thick Wall Atmospheric Exhaust, 16″ Riveted Steel	3½″ and under C. I. Screwed—Std. 4″ and over C. I. Flanged—Std.	C. I. Screwed—F. & D. to Amer. Std. for125 Lb.	Same as for 125 Lb.		1/16″ Thick	2″ and under M. I. Br. Seat (Use Flanges over 2″)	As Above	As Abovefor Steam Piping	One Coat Red Lead and Oil
Boiler Feed Water 300 Lb.	Ex. Hy. Seamless Steel	Ex. Hy. C. S. Flanged	Ex. Hy. C. S.—Screwed and Weld—F. & D. to Amer. Std. for 250 Lb.	Ex. Hy. C. S. Flanged—O. S. & Y. Bronze Mounted	Gate Globe	1/16″ Thick	None	Studs and Stud Bolts—Alloy Steel—Med. Carbon Steel Nuts—Hex.	1½″ Thick Air Cell Sectional Covering—Canvas Jacket—Painted	One Coat Red Lead and Oil
Water—125 Lb.	10″ and over, C. I. in Tunnel, Flanged in Pump House and Power House 8″ and under Full Wt. Steel	3″ and under C. I. Screwed—Std. 3½″ and over C. I. Flanged—Std.	10″ and over—Flg's Cast on Pipe 8″ and under C. I. Screwed All F. & D. to Amer. Std. for 125 Lb.	3½″ and under—Brass—Screwed I. S. Rising Stem 3½″ and over O. S. & Y. C. I. Body Bronze Trim—Flanged	Globe Gate Globe Gate Check.	1/16″ Thick	2″ and under M. I. Br. Seat	Std. Steel Bolts—Sq. Hd. Hex. Nuts	Hair Felt—Special for 16″ Water Pipes	6″ Pipes—One Coat Asphalt Paint Steel Pipes One Coat Red Lead and Oil
Boiler Blow-Off 250 Lb.	Ex. Hy. Seamless to B. O. Valve Ex. Hy. Steel Pipe from B. O. Valve to Tank	Ex. Hy. C. S. Flanged	Boiler to B. O. Valve—Forged Steel—Screwed and Weld. B. O. Valve to Tank—C. S. Screwed—F. & D. to Amer. Std. for 250 Lb.	Tandem B. O. Valves Ex. Hy. C. S. Body—Flanged Monel Metal Trim	Seatless	1/16″ Thick	None	Alloy Steel Stud Bolts—Steel Hex. Nuts	None	One Coat Red Lead and Oil
Compressed Air Piping	Std. Wt. Steel	C. I. Screwed—Std.	None	Brass—Screwed	Globe Gate	None	M. I. Brass Seat	None	None	One Coat Red Lead and Oil
Lubricating and Operating Oil Piping	Std. Wt. Steel	Brass—Screwed	None	Brass—Screwed	Globe	None	M. I. Brass Seat	None	None	One Coat Red Lead and Oil

All pipes except 16 in. O. D. water mains to have adjustable pipe supports. 16 in. O. D. pipes to set in C. I. saddles.

Handling Condensate

The condensate from the turbines and other apparatus is piped to a condensate tank placed on the fan floor. Piping is arranged to supply the boiler feed pump through the deaerating heater from this condensate tank; thus, when the make-up plus the condensate exceeds the demand of the boiler feed pump, the piping is arranged so that the excess water may overflow from the condensate tank to a storage tank placed in the basement. In the event of low water in the condensate tank a float switch operates the hot water pump and the supply is replenished from the storage tank. Sump pumps are employed to take care of drainage, the discharge from these pumps being piped to the return line from the condensers.

All piping in the power house is arranged so that valves and accessories are readily operated. Expansion in the piping is taken care of by means of expansion bends, except in the pipe tunnel where long lengths of steam pipe demand a slip type expansion joint.

Reliability of Equipment Imperative

A complete piping schedule giving a summary of specifications for the piping and accessories for this plant is given in Table 1. One of the factors that governed the choice of material was the isolated location of the plant. Churchill lies approximately 800 miles north of Winnipeg and approximately 500 miles from The Pas. It was rigidly set out in the specifications prepared by the consulting engineers for the Department of Railways and Canals that no equipment was to be installed that had not passed the experimental stage and found absolutely reliable. A shut-down due to failure of any apparatus during the wheat shipping period would cause considerable loss.

To Operate Three Months per Year

For the present the power plant will be operated only during the period when shipping through the Hudson Strait is possible, approximately three months during the fall of the year. This had some bearing in the choice of pipe covering, in that the insulation that gives greatest Btu saving is not the most economical when the investment in pipe covering is considered for this short period of operation.

All apparatus, pump bases, piping, etc., where water may collect is arranged so that it may be drained dry when closing the plant after the shipping period, in order to prevent water collecting and freezing. Sloping of pipe in certain cases is necessary to accomplish this. With winter temperatures of 50 F below zero not uncommon it makes this precaution necessary. It was due to these low temperatures and also that the frost leaves the ground to a depth of only eighteen inches during the summer months that special consideration was given to the design of the pipe tunnels. Steam heating of the cold water pipes is also necessary to prevent freezing.

If, at a later date, this plant operates through the winter it may be necessary to provide steam coils around the pump suction piping in the wells, and the system of sluice gates in the wells will have to be operated so that the warmer water returning from the condenser will assist in preventing freezing of water in the intake well.

C. D. Howe & Co., Port Arthur, were the consulting engineers for the Department of Railways and Canals; Babcock-Wilcox and Goldie-McCulloch, Limited, Galt, were the contractors for power plant equipment and power plant piping and design, and Cotter Bros. Limited, Winnipeg, were the contractors for heating, piping and plumbing and also supplied material and labor for power plant piping.

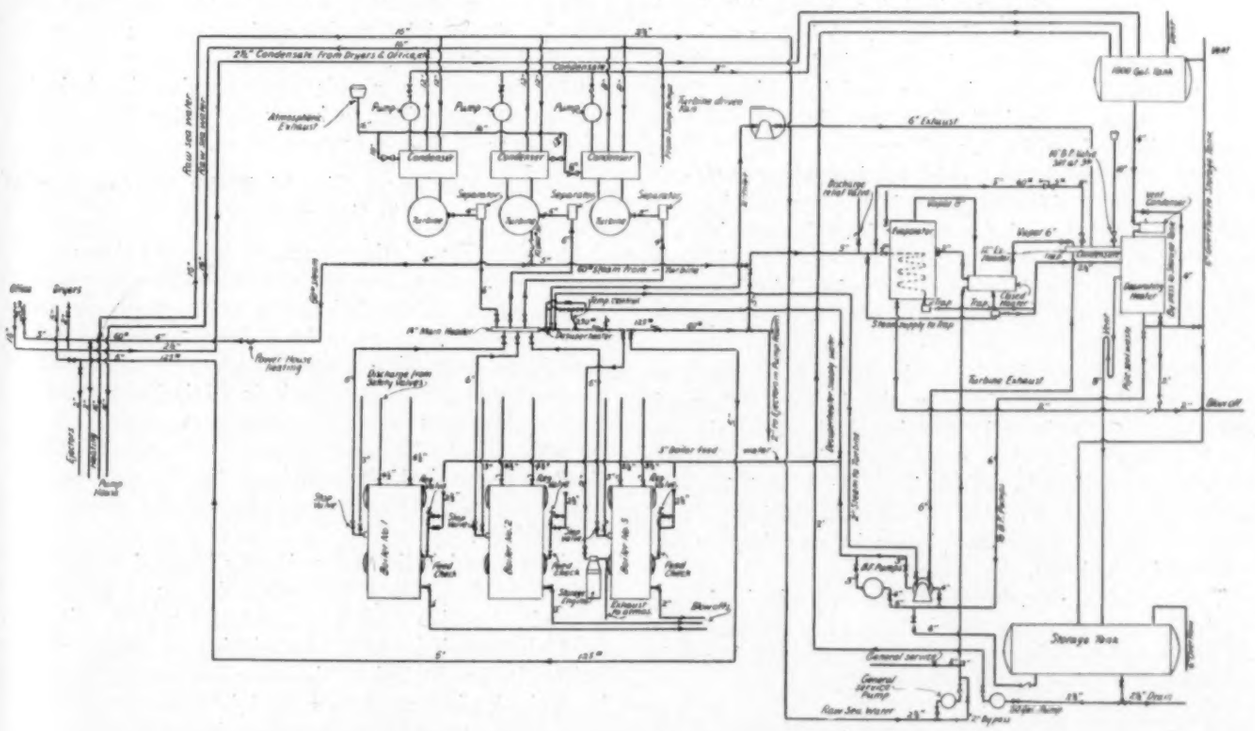

FIG. 6—COMPLETE DIAGRAM OF THE POWER PIPING

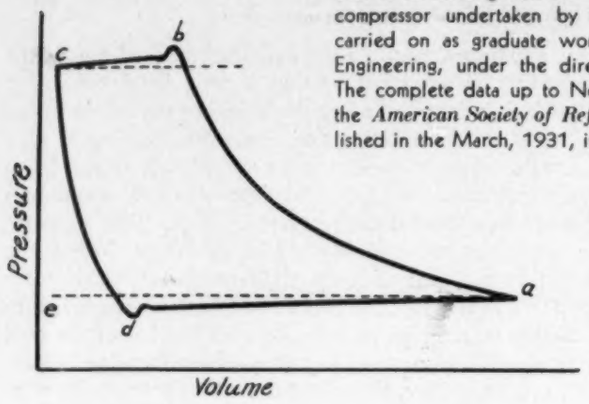

FIG. 1—INDICATOR CARD, SHOWING EFFECT OF WIRE DRAWING AND VALVE ACTION

In an air conditioning plant or any refrigerating plant using the compression system, the compressor is of great importance, whether it is of the reciprocating, centrifugal, or rotative type. This paper deals only with the reciprocating type and points out the factors to be observed for good compressor design and operation. In order to obtain maximum economy, particular attention should be given to volumetric efficiency, since it is a very important factor.

The experimental data included herein were obtained as a portion of an investigation on the volumetric efficiency of an ammonia compressor undertaken by the authors. This investigation was carried on as graduate work, in the Department of Mechanical Engineering, under the direction of Professor H. J. Macintire. The complete data up to November 1, 1930 were submitted to the *American Society of Refrigerating Engineers* and were published in the March, 1931, issue of *Refrigerating Engineering.*

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and
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Improved Compressor Operation a Road to Air Conditioning Economy

THE refrigeration required for air conditioning is becoming greater and greater. A large portion of the total refrigeration in the United States today is required for air conditioning, as used in various places and processes. The indications are that in the near future this work will grow still greater.

In air conditioning or refrigeration where ammonia is used as the working medium, in the compression system, a thorough knowledge of the losses[1] occurring and how to prevent them is necessary for economical operation. Some of these losses may be reduced by improved design but the *majority of them may be eliminated by careful operation.*

Practically the entire losses in capacity of the compressor may be accounted for by what is called volumetric efficiency. The writers, in studying the losses of the ammonia compressor, find no standard or adopted definitions relative to volumetric efficiencies. As a probable consensus of writers the term "volumetric efficiency" used to cover all the compressor losses is a total or overall volumetric efficiency.

The apparent or volumetric efficiency obtained from the indicator card is one that is frequently referred to as the volumetric efficiency of a compressor, but in reality it represents the effect of clearance alone.

The real volumetric efficiency is the one which may only be obtained by testing and represents losses due to superheating, leakage, etc.

It is the purpose of this paper to give the designing and operating engineer a better understanding of these efficiencies and how to improve them.

Operating Engineers Should Make Proper Use of Indicator Cards

The function of the compressor is well known. It simply serves to remove the gas from the evaporator to the condenser. This obviously requires that work shall be done on the gas. In order to know that this work is done in as economical a manner as the construction of the compressor will permit, it is advisable to take indicator diagrams from the compressor cylinder frequently. These diagrams should be compared with an ideal or theoretical diagram in order to learn to what extent any factor or factors influence the operation of the machine.

These actual indicator cards may be used to good advantage to show many mechanical defects of a compressor, all of which may affect the volumetric efficiency. Such characteristics as valves sticking or leaking, piston leaks, wiredrawing, pipe friction and clearance losses will be noticed when an actual card is compared with the theoretical one under similar operating conditions.

There are some factors that a diagram will not show,

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[1] The term "losses" as used in this paper refers to those losses that affect the capacity of the compressor.

however. One factor, for instance, which is of great importance and which is not shown by the card is the quality of the suction vapor entering the compressor cylinder. This fact is of vital importance, but is neglected in many plants. The presence of liquid in the cylinder tends to lower the volumetric efficiency.

The indicator card reveals much to the operating engineer, if properly interpreted. The fluctuations in the discharge, at *b*, Fig. 1, and the suction at *d*, show the additional pressure required to actually open the valve. The space below the suction pressure line indicates the force required to push the gas through the valve due to small valve area, causing wire drawing.

Proper Clearance Should Be Maintained

Two points a designer of ammonia compressors must lay most stress on are, the stuffing box and clearance in the cylinder. A slow speed machine, in order to be efficient, must have very little clearance on account of the expansion of the gas. This re-expansion increases with an increase in the amount of clearance in the cylinder.

The clearance in the compressor is not a fixed quantity, but changes with the natural wear of cranks and cross-head. For this reason it is advisable to provide clearance marks on the guides and cross-head of the compressor. These marks will aid in the matter of equalizing the clearance at the end of the cylinder. They may also aid in taking up the wear of the bearings.

Proper Packing Important

Another source of trouble may arise if the packing between the head of any cylinder is too thick. This gasket should be as thin as conditions will permit. The head clearance usually should not exceed 1/32 of an inch for a slow speed machine and 1/16 of an inch for a high speed machine.

The greater the compression ratio the greater the loss due to clearance, but with the same clearance the loss is less in a long stroke compressor. If the compression ratio becomes relatively high, the compression should be accomplished in two or more stages.

Apparent Volumetric Efficiency

The effect of clearance upon the volumetric efficiency may be determined as soon as the amount of clearance is known. The fact that there is clearance does not affect the work of compression; it affects only the capacity. In the ammonia compressor the ammonia gas remaining in the clearance space expands on the return stroke of the piston along the curve *C D*, Fig. 2. The efficiency as a result of the loss due to the re-expansion only of the gas in the clearance space is called the apparent volumetric efficiency, E_A.

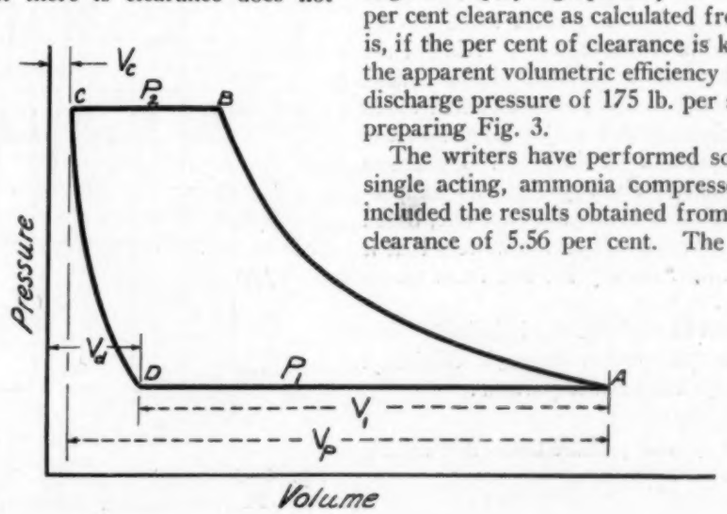


FIG. 2—INDICATOR CARD, SHOWING EFFECT OF CLEARANCE

In Fig. 2, the volume displaced by the piston is represented by V_p, the clearance by V_c, the volume after re-expansion by V_d and the volume of ammonia gas entering by V_1. E_A, the apparent volumetric efficiency is the ratio of V_1 to V_p. These values may be scaled from the indicator card but this practice is not always satisfactory as the results are often misleading.

It is rather easy to get high values of efficiency with mechanically operated intake valves. For instance, by opening the intake at the inner end of the stroke, allowing the gas compressed in the clearance space to flow into the intake pipe to obtain a vertical drop from the discharge to intake pressure, this may be done as far as the indicator diagram is concerned. Also, from Fig. 2, it can be seen that the longer V_1 becomes, the greater the volumetric efficiency. Reducing the clearance space will increase V_1.

From Fig. 2 we have:

$$V_1 = V_p + V_c - V_d$$

$$V_d = V_c \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

Let $m = \dfrac{V_c}{V_p}$

Then $V_1 = V_p + m V_p - m V_p \left(\dfrac{P_2}{P_1}\right)^{\frac{1}{n}}$

$$V_1 = V_p \left[1 + m - m \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\right] \qquad (1)$$

But $\dfrac{V_1}{V_p} = E_A$

$$E_A = 1 + m - m \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} \qquad (2)$$

Fig. 3 displays graphically the effect of varying the per cent clearance as calculated from Equation (2). That is, if the per cent of clearance is known one can calculate the apparent volumetric efficiency from Equation (2). A discharge pressure of 175 lb. per sq. in. abs. was used in preparing Fig. 3.

The writers have performed some tests on a vertical, single acting, ammonia compressor and in Fig. 3 have included the results obtained from a compressor having a clearance of 5.56 per cent. The compressor was oper-

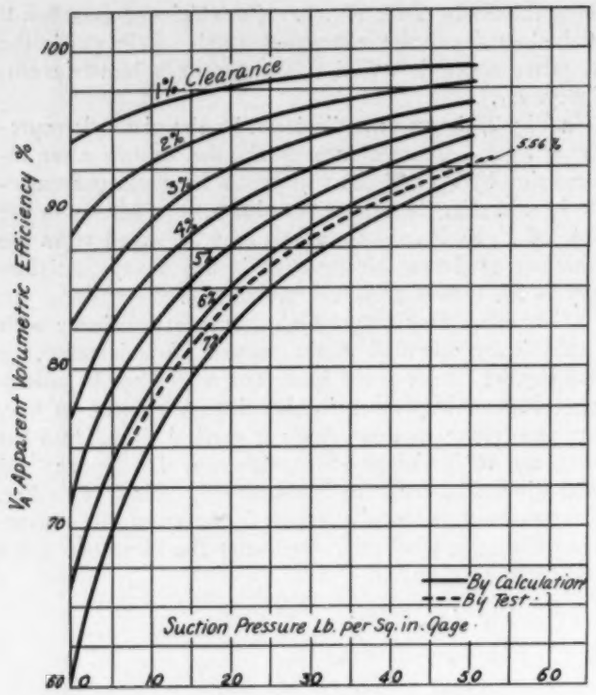


FIG. 3—EFFECT OF VARYING THE PER CENT CLEARANCE UPON THE VOLUMETRIC EFFICIENCY

ated at the five suction pressures noted and at a discharge pressure of 175 lb. per sq. in. abs. It will be noted that the test curve follows the calculated one quite closely; however, it is about 1 to 3 per cent low. This is within the limits of experimental accuracy.

Real Volumetric Efficiency

The real volumetric efficiency of a compressor as defined earlier in this paper represents the losses due to superheating, leakage, etc. In order to evaluate this efficiency it becomes necessary to actually test the machine. In Fig. 4, the results of such tests, performed on a vertical, single acting, ammonia compressor may be observed. These curves represent the losses due to the superheating of the gas upon entering the cylinder, wire-drawing, leakage past the piston and valves, etc.

In Fig. 5 the real volumetric efficiency as obtained from several different sources is plotted against absolute discharge pressure. There does not seem to be a very close agreement with the three methods of arriving at the real volumetric efficiency of a vertical, single acting, ammonia compressor. This seems to indicate that much remains to be done to clear up this subject of volumetric efficiency as to size, type, etc. This graph also shows that the Voorhees formula does not give results comparable with tests. This formula is an empirical one, based on the work of Denton. It appears to the writers that it applies to only one machine, namely the one from whose data it was derived.

It is a difficult problem to evaluate the individual components which make up the real volumetric efficiency. The following are perhaps the most important:

(1) Superheating effect
(2) A reduction of the cylinder pressure below that existing in the suction line
(3) Speed
(4) Valves
(5) Jacket water

In the following paragraphs a discussion of each of these factors will be made so that the operating engineer may make adjustments to attain higher capacity with the same input to the machine.

Superheating Effect

To obtain maximum refrigeration from a machine, the gas should come back to it at such quality that at the end of the compression it will just be saturated at the existing head pressure. This will tend to make the cycle of operation approach the Carnot cycle and means that the compressor would work under wet compression. Should the gas return to the machine in a superheated condition it would be termed dry compression. Theory shows a gain of economy in favor of wet compression.

Under actual operating conditions one can not operate a machine satisfactorily under these conditions. For actual operating conditions experiments by Prof. Schroeter show that a slight degree of superheat is advantageous to performance. This does not agree with

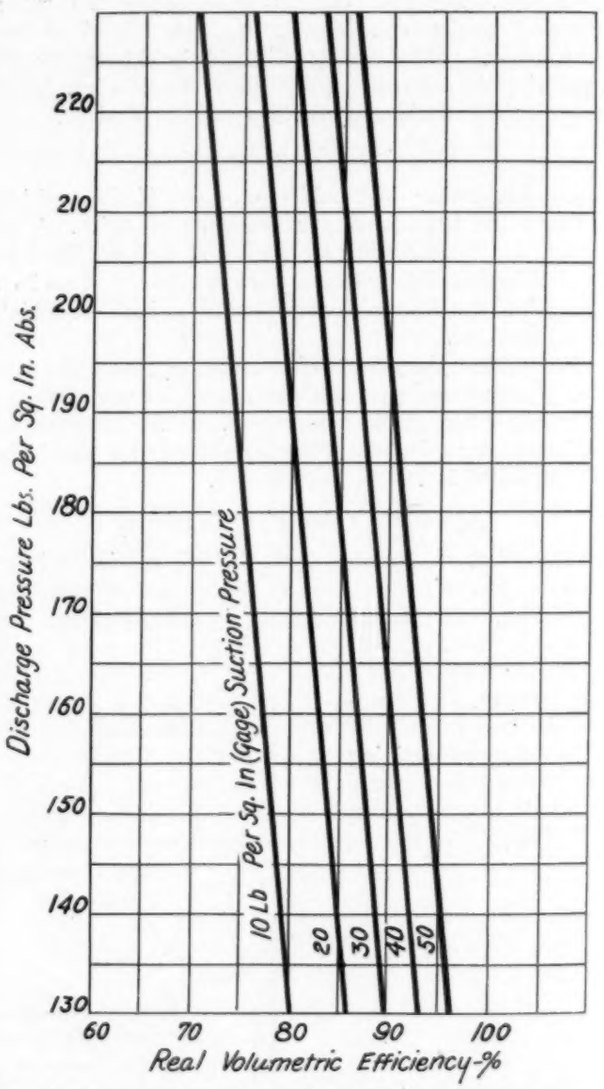


FIG. 4—AUTHORS' TESTS FOR REAL VOLUMETRIC EFFICIENCY PLOTTED AGAINST ABSOLUTE DISCHARGE PRESSURE

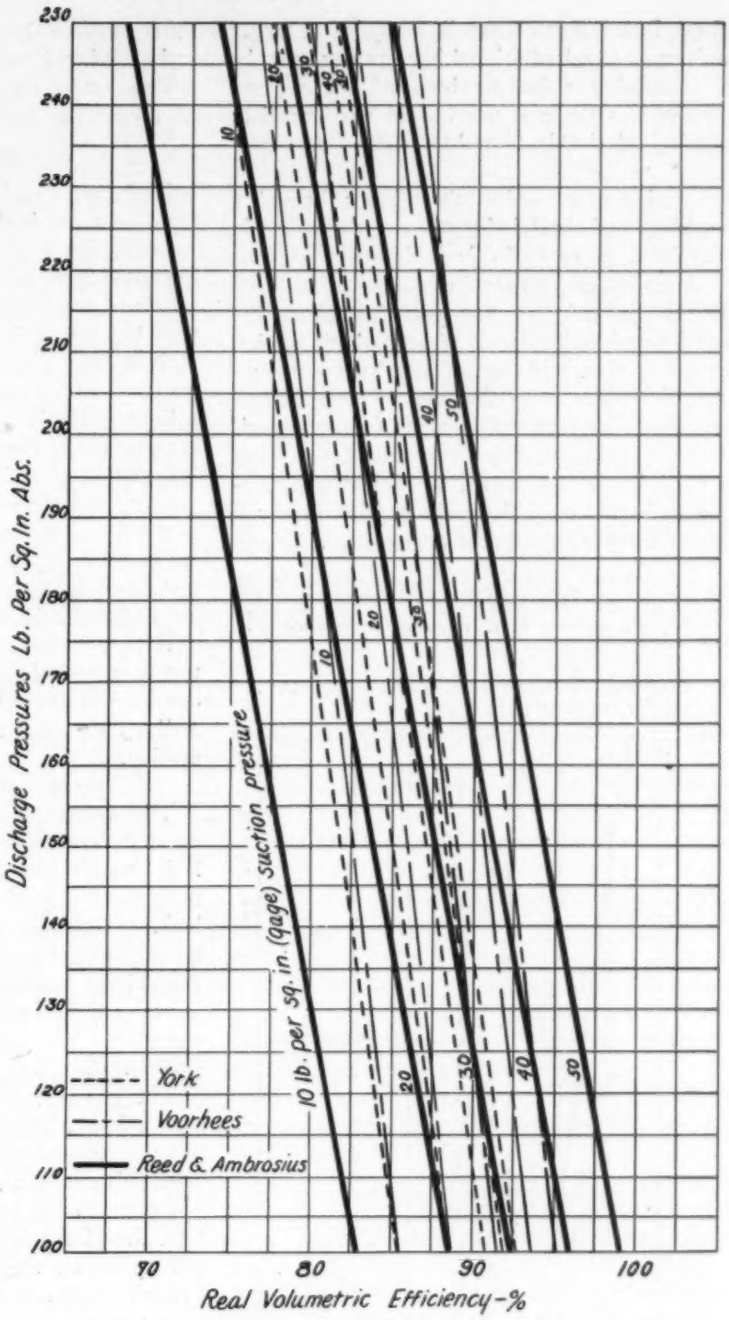


FIG. 5—COMPARISON OF AUTHORS' TESTS WITH RESULTS OF THE VOORHEES FORMULA AND THE YORK TESTS

theory, but in practice true adiabatic compression is not obtained and the indications are that some superheat helps the performance by decreasing the heat exchange between the cylinder and gas. Superheated vapor behaves somewhat like a gas, in that it gives off less heat by conduction than the saturated vapor.

There is a greater loss in capacity with the suction vapor superheated a few degrees than when a small quantity of liquid enters the cylinder. However, it must be remembered that on account of the shallow construction of the stuffing box on dry compressors, it is not practical to operate even with the suction gas in a saturated condition, for it is almost impossible to maintain a saturated condition for any length of time.

This is especially true with varying load conditions. The consensus of opinion seems to be that it is better to tolerate a few degrees of superheat in the suction gas. The volumetric efficiency is greatest for this condition, considering practical limitations.

The oil injection compressor has been used with varied success to counteract the superheating effect. In this machine refrigerated oil is circulated through the cylinder by a small pump; another object of this oil is to reduce the injurious effects of clearance or at least lessen them materially. This also aids in lubrication and in making the piston tight.

Reduction of Cylinder Pressure Below that in Suction Line

Upon examining an indicator diagram it will be found that the pressure in the cylinder is materially lower than that existing in the evaporator. The loss arising from this has already been mentioned.

In the first place, the valves—as a rule—are automatic. On account of this, it requires a difference in pressure to raise the valves from their seats and support them against inertia and spring tension. The power required to open the valves is practically constant while the loss due to pipe line friction depends on the velocity, pipe size and properties of the gas. Much work is also lost when the resistance in the suction and discharge lines is too great. This may be due to over-weighting of the valves, or dirty and obstructed piping.

Economical Pipe Sizes Should Be Used

Frictional losses are detrimental to compressor efficiency; they are directly dependent upon pressure drop. If a large pressure drop occurs in the suction line of a compressor, there is a loss in refrigeration. A pressure loss in the discharge line means an increase in the horsepower to produce the refrigeration desired. In order to

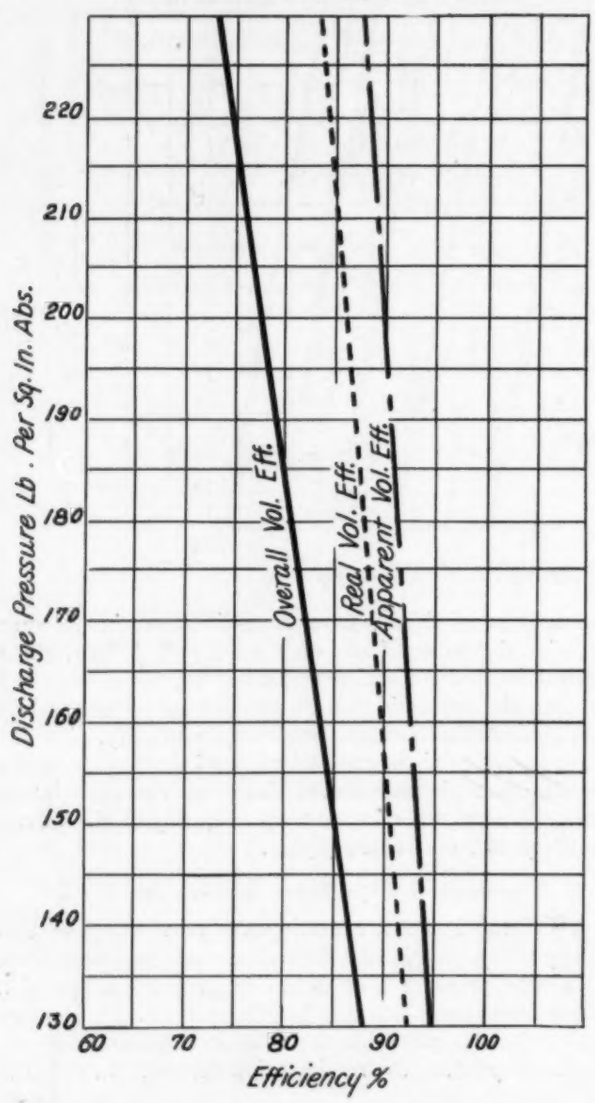
overcome these losses there must be supplied a larger cylinder displacement and additional horsepower must be furnished. Increasing either or both of these items means an increase in the cost per ton of refrigeration. Economically it is desirable to reduce these items to a minimum.

The piping engineer may reduce these items through a thorough knowledge of the frictional losses occurring in ammonia pipe lines. A part of this field was covered by the writers[2] about a year ago. Having ammonia analyzed according to the dimensionless homogeneity theory the designer may determine the size of pipe for both suction and discharge line that will permit the least pressure drop. In the discussion upon the flow of superheated ammonia the writers gave an explanation of their experimental data and a typical problem was solved.

The matter of insulation should be considered carefully. It is very important to have the suction line well insulated, from the evaporator to the compressor. Everything possible should be done to be sure that the greatest weight of gas will get into the cylinder and be compressed under the given suction conditions. The density of the suction gas should be kept high.

[2] "The Flow of Superheated Ammonia in One Inch, Extra Heavy, Black, Steel Pipe," by John C. Reed and Edgar E. Ambrosius. HEATING, PIPING AND AIR CONDITIONING, Feb. 1930. Vol. 2. No. 2. An article on this same subject also appeared in *Refrigerating Engineering*, Feb. 1930. Vol. 19. No. 2.

FIG. 6—COMPARISON OF THE APPARENT, REAL AND OVERALL VOLUMETRIC EFFICIENCY



Speed of Compressor

The capacity of a refrigerating system, if of the compression type, is dependent upon the speed of the compressor. Just what effect the speed really has on the compressor is something that is rather doubtful in the minds of many engineers. There seems to be no question but what the capacity of a compressor increases with speed. Test data show that the capacity of a compressor changes approximately with the speed, or is a linear function of it. This can easily be true if the compressor is properly designed and the valves are sufficiently large. However, this being true, then under similar conditions the volumetric efficiency must be independent of the speed. This phenomenon can readily be understood, since ammonia is not a very active liquid and, therefore, as the piston speed increases the re-expansion in the cylinder decreases.

Before these assertions are absolutely proven more test data must be available on various types of machines operated under different conditions. However, this reasoning has some merit and should receive due consideration.

Valves

The success of compressor operation often depends upon the condition of the piston, rings and valves, since leaky valves and piston will give poor economy. After long usage the piston and valves are almost certain to leak. If a leak is greater in the discharge valve, the compression curve tends to become straight and steep. Should the leak be in the suction valve, the compression curve will be rather flat. This can all be verified by comparing the actual card with the true *adiabatic* and *isothermal*.

The gas will leak from one side to the other if the piston leaks. This will cause a longer interval of time for the suction pressure to reach the normal. The suction valve will not open until the speed of the piston increases to the point where the volume of gas leaking past the piston is no longer sufficient in quantity to fill the space on the suction side; then the pressure decreases gradually to the point where the suction valve opens.

Jacket Water

The amount of jacket water does not materially affect the volumetric efficiency of the compressor. Modern makes of double acting compressors are provided with water jackets; this, however, is not always adequate in order to keep the temperature of discharge down. Since it is high discharge temperature that affects the re-expansion so much, liquid ammonia is often fed directly into the suction line.

The indicator card shows that the latter part of the actual compression curve deviates from the adiabatic and approaches the isothermal. This performance is in part attributed to the water jacket. However, the adiabatic curve will remain the standard of comparison, unless some method is provided by means of which the cylinders will be completely surrounded by a cooling medium that will bring the compression curve nearer to

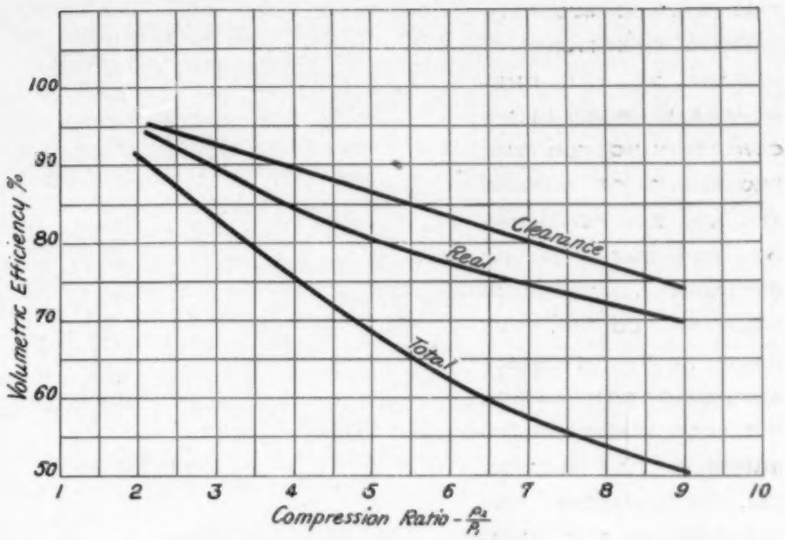


FIG. 7—VOLUMETRIC EFFICIENCIES VS. COMPRESSION RATIOS

the isothermal. This is rather doubtful, since the piston, etc., are poor conductors of heat. The partial solution is stage compression.

Overall Volumetric Efficiency

The overall volumetric efficiency is the ratio of the actual amount of vapor compressed and delivered, expressed in cubic feet per minute at intake pressure and temperature, to the compressor displacement. This overall efficiency is the result or product of the apparent and the real volumetric efficiency. For example, if the apparent volumetric efficiency is 50 per cent and the real volumetric efficiency is 50 per cent, the overall volumetric efficiency would be the product of the two or 25 per cent. This is also represented by $E_T = E_A \times E_R$.

Fig. 6 shows the apparent, real, and overall volumetric efficiency, at a constant suction pressure, of a vertical, single acting compressor having a clearance of 5.56 per cent, as determined by the writers. This graph is included chiefly to point out the difference in these three efficiencies.

In the ultimate analysis this overall efficiency is the one which is of greatest interest. As it is composed of the apparent and the real volumetric efficiency it may be improved only by improving them, through careful operation. The clearance should be reduced to a minimum, and all leakage, etc., prevented. This will make E_A and E_R higher, which in turn tends to increase E_T. It is the overall (E_T) volumetric efficiency which is of vital concern, the capacity.

A study was also made of the relation between the volumetric efficiency and compression ratios. It would be expected that as the compression ratio approaches unity, the volumetric efficiency approaches 100 per cent. The experimental results of this relationship are shown in Fig. 7. The variation of the apparent, real, and overall volumetric efficiencies are clearly brought out and this graph substantiates the suspicion that the compression ratio approaches unity as the volumetric efficiency approaches 100 per cent.

Conclusions

As a result of the writers' study of volumetric efficiency, it is felt that any conclusions drawn must be of a two-fold nature. That is, there are certain conclusions that concern design or the theoretical side of this problem, and there are certain conclusions that concern the operator or the practical side of the problem. The following conclusions may be considered as a result of studying the problem more or less theoretically:

(1) From the bibliography studied, which was very complete, it is felt that the information relative to the volumetric efficiency of an ammonia compressor is very meager.

(2) Up to the present time, sufficient data have not been accumulated to warrant the selection of an empirical formula for arriving at the volumetric efficiency based on tests.

(3) Much more valuable information could be obtained concerning volumetric efficiency if a satisfactory means were devised for finding the temperature at the beginning or at both the beginning and end of compression.

(4) Valves and parts should be designed carefully and especially be of ample size.

(5) All losses that might occur in operation should receive consideration in design.

(6) Any formula to be used for determining the actual displacement of a compressor necessary to produce certain results must contain a factor allowing for the volumetric efficiency properly.

(7) The fact that the tests of Denton [3] and those of Horne and Oakley[4] check the authors' values, helps to substantiate the writers' tests.

(8) This subject should receive serious attention by everyone who has occasion to use the compression system of refrigeration.

Of course, there is an overlapping of these conclusions in that some of them apply to both design and operation. The following conclusions may be considered to apply more specifically to the operating engineer.

(1) It is important to prevent all leakage in the compressor and plant.

(2) The indicator card should not be depended upon entirely to give an accurate record of performance.

(3) The importance of running at low compression ratios should be realized.

(4) The value of insulation should be considered.

(5) Great care should be exercised in repair and upkeep. For example, the replacing of a head gasket with a new one much thicker than the old increases the clearance volume materially.

(6) The operating engineer may profit by some of the suggestions set forth in this and other such papers and thereby reduce the compressor losses, increase its efficiency and increase its capacity. This all results in a reduction in the cost per ton, an increase in tonnage and likewise an increase in net profits.

[3] "Performance of a 75-Ton Refrigerating Machine of the Ammonia Compressor Type," J. C. Denton. *Trans. A. S. M. E.* Vol. XII. 1890.
[4] "Test of a Two Stage, Double Inlet Ammonia Compressor," G. A. Horne and A. W. Oakley. *Refrigerating Engineering*, 19, 6, June, 1930.

THIS ARTICLE SUGGESTS SEVERAL QUESTIONS IN REGARD TO THE PHENOMENA INVOLVED IN AIR CONDITIONING FOR THE PROCESSING OF MATERIALS AND AIR CONDITIONING FOR THE COMFORT OF HUMANS. THERE IS MUCH WE DO NOT YET KNOW ABOUT "CAUSES." ACCORDING TO THE AUTHOR "IT DOES NOT SEEM IMPOSSIBLE THAT ELECTRICAL PREDICAMENT MAY ACTUALLY BE THE PARAMOUNT CONSIDERATION IN ANY FORM OF AIR CONDITIONING ENDEAVOR."

By Edgar S. Miller*

Air Conditioning—A Discussion of Some of the Phenomena Involved

A Relation Between Static Electricity and Moisture Regain?

IN THE lexicon of air conditioning engineering the term *regain* is often considered equivalent to *moisture content*. This is sometimes quite misleading to the physicist, who usually thinks of the word as having to do with moisture lost and regained—or at least to moisture gained by a body from some exterior source.

There never was any real reason for this general use of the term, and the profession could very well afford to change its ideas and restrict the usage to the bounds of legitimacy. For whether the fact is generally known or generally recognized or not, the fundamentals of air conditioning draw a sharp line between the conditions responsible for a loss or gain of moisture by hygroscopic bodies.

Meaning of "Hygroscopic"

Since the writer's use of the term *hygroscopic* in connection with the characteristics of certain substances has been questioned in the same manner, it may be well to defend it before proceeding with the challenge of the use of *regain*. Hygroscopic means, literally, "water that can be viewed," and under a strict interpretation a piece of cold glass or metal might be called a "hygroscopic" body. But since the botanist has given to the word a special meaning when he applies it to the properties possessed by vegetable tissues, by virtue of which moisture is absorbed or given off, it doesn't appear that its employment in connection with bodies that are not only capable of receiving on their surfaces water condensed from the atmosphere, but are also capable of "absorbing" this moisture and thus holding it until a combination of circumstances forces re-evaporation, is nearly as illogical as is use of *regain* for *moisture content*.

Moisture Regain and Static Electricity

Recently an air conditioning engineer of note stated that "the industrial application of air conditioning to any field revolves principally around moisture 'regain'

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and static electricity." He voiced the opinion that there is probably more connection between these two factors than the air conditioning engineer has suspected. Physicists are beginning to suspect a definite connection between heat and electro-magnetic force, as is evidenced by the expressed thought of W. H. Carrier, in a paper read before the *American Society of Refrigerating Engineers* in 1927.

Perhaps there *is* an all-important connection between the factors known as *static electricity* and *regain,* if we consider the term *regain* to mean the phenomena attending the transformation of sensible to latent heat, and vice versa, involved in moisture loss and moisture gain by "hygroscopic" bodies, and perhaps this connection is significant in fields other than industrial air conditioning.

Disabusing ourselves of the misleading term *regain,* and considering every problem concerned with a gain or loss of moisture by atmospheric air as due primarily to heat metamorphosis, we can readily identify all changes in moisture content by hygroscopic bodies with some manifestation of heat. It is not impossible that in doing so we may have also exposed the connection between electricity in some form—even though we are unable at the present time to identify it—and this heat.

"pH Concentration"

In connection with the processing of certain cereals and their products—such as the milling of flour and the baking of bread—it is well established that electrical manifestations play an important part. A few years ago the cereal chemist conceived the notion that the measurement of the positive hydrogen factor would give him a key with which he could expose the secrets of colloidal chemistry, as it pertains to the behavior of cereal products.

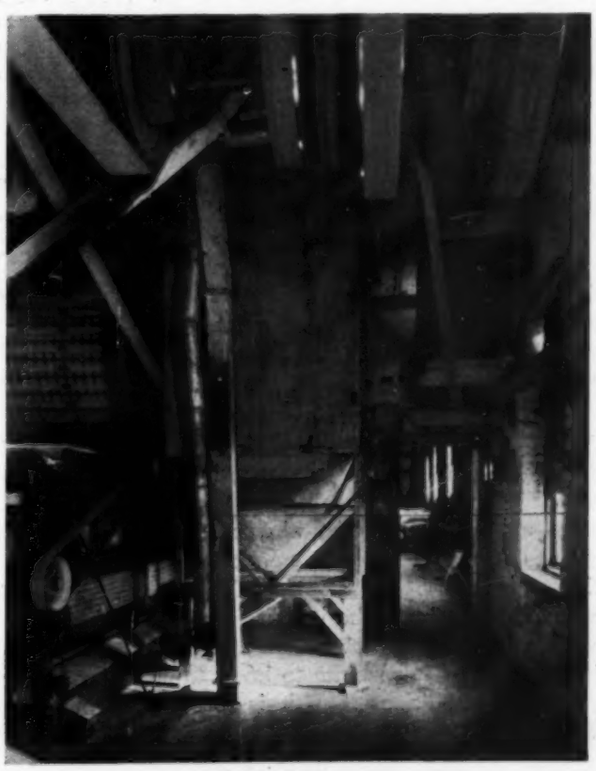
A vast amount of work was done, and some erroneous conclusions may have been drawn. Certainly some extravagant claims were made. It was assumed that the physical properties of the protein colloids could be altered at will by an increase or diminution of *acid activity,* measured by the excess of positively-electrified hydrogen atoms over those negatively electrified. The term, *pH,* expressed as the negative logarithm of the ratio of positively electrified hydrogen atoms to the whole number—the *pH concentration*—became quite common. Only a few—even among the chemists themselves—knew exactly what it meant. And though it would not be truth to say that the behavior of acids, as manifested by the ionization of hydrogen atoms, is not readily understood by a large number of physicists and chemists, one might say with little fear of contradiction that very little is known concerning the significance of these manifestations.

It does not seem to occur to many engaged in air conditioning that the main purpose is actually that of controlling the forces responsible for the ultimate conditions existing in and on the surfaces of the bodies affected. The temperature and humidity existent within a few inches of a body may give little or no evidence of these conditions, or—what is perhaps of greater importance—of the phenomena that have brought these conditions about.

Important to Know Cause of Temperature and Humidity Conditions

This may be roughly illustrated by an example. In an industrial plant processing hygroscopic materials containing free water, the dew point in the rooms heated but not humidified is invariably higher than that of the out-of-doors. The absolute humidity of the room air is increased by the evaporation of water from the materials. Suppose, with an outdoor temperature of zero the room conditions are dry bulb, 70 deg; wet bulb, 56 deg; dew point, 44.5 deg; relative humidity, 40 per cent. Unaware of the significance of the mysterious happenings that result in an increase of moisture in the room air approximately 3 grains per cubic foot, and having evidence that the conditions obtaining are not ideal for the processing, we might conclude that the 40 per cent relative humidity is too low.

If we begin to introduce highly humidified air into the room it is obvious that as the evaporative power of the air is decreased by vaporization in the humidifier, vaporization from the materials will decrease. Thus we



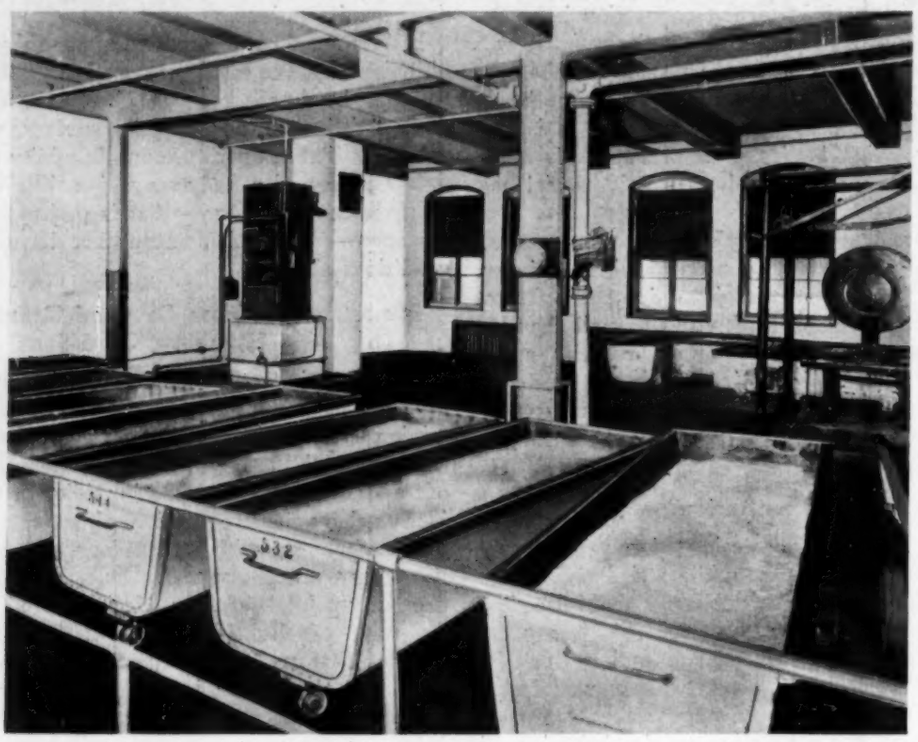
Cloth Dust Filler (in Left Center) Is Discharging Air. The Vapor Pressure of the Moisture in This Air Profoundly Affects the Absolute Humidity of the Room Air, and This, in Turn, Influences Evaporation from Materials, and Consequently Their Temperatures

might proceed with our humidification to a considerable extent and not alter the dry bulb, wet bulb and dew point, as indicated by room measurements, in the slightest degree, and still produce a pronounced change in the conditions affecting the processing. If we were to conclude that the first-mentioned conditions indicated a too-low percentage of relative humidity, and were to increase the dew point of the room air to about 55 degrees, or a relative humidity of about 60 per cent with a dry bulb of 70, we might still more seriously affect the processing by preventing evaporation from the materials —for if these materials were receiving heat from an exterior source, under the conditions of a low dew point in the outdoor air they would be losing much of this heat by transition from a sensible to a latent state, their temperatures being correspondingly reduced while the dew point of the used air was increased.

Perhaps no one can do more than guess at the nature of the electrical phenomena accompanying this heat-and-moisture metamorphosis—or inhibited by its absence. The point is, it is not safe to estimate the probable effects on materials of altering atmospheric conditions in the room unless one goes much deeper than the indications of conventional measuring instruments located in the room. In other words, it is of less importance to know the temperature and humidity conditions than it is to know "how they got that way."

"Stuffy" Sensation in Crowded Rooms

In the field of air conditioning for comfort these phenomena are of no less importance. Scientific research has exploded the old notion that a "stuffy" sensation is certain evidence of a condition in which the body suffers from an excess of carbon dioxide in the air breathed. Yet human beings do suffer in over-crowded rooms, and the suffering is relieved by the introduction of "fresh air."



THE SURFACE OF THE DOUGH IN THE TROUGHS IS AFFECTED BY THE EVAPORATION OF MOISTURE AND THE PASSING OF CARBON DIOXIDE INTO THE AIR. IT IS THOUGHT THAT THE PHENOMENON IS ACCOMPANIED BY ELECTRICAL MANIFESTATIONS WHICH ARE POSSIBLY THE SIGNIFICANT FACTORS TO BE CONSIDERED IN CONDITIONING THE AIR IN THE DOUGH-ROOM. THE APPARATUS IN THE LEFT BACKGROUND CONTROLS THE TEMPERATURE AND RELATIVE HUMIDITY OF THE AIR BEING DIFFUSED INTO THE ROOM

If an excess of carbon dioxide is not responsible for that uncomfortable feeling one gets when packed in among other perspiring martyrs in an unconditioned theater or auditorium, what is responsible? An excessive relative humidity? Perhaps. But it is true that a lone occupant of a large room may experience a great deal of discomfort when surrounded by air comparable in moisture content to that in Death Valley at the height of summer.

Causes of Discomfort in High Humidities

The animal body differs considerably from those referred to as "hygroscopic." It does not "absorb" moisture from air, but it does freely contribute moisture which will, under certain conditions, vaporize into the air, the rate of evaporation being governed by the evaporative capacity and the actual quantity of air available. Thus some evaporation would occur from the skin of a nude human being if surrounded by air at 80 deg. and

100 per cent relative humidity; for the body would be warmer than the air, and would therefore contribute some heat to the perspiration on the skin. This would increase the vapor pressure of the perspiration above that of the air, and until equilibrium was reached, vapor —and consequently heat—would flow from the higher to the lower level. But the amount would be quite small; and as the sweat glands, almost failing in their normal function of disposing of bodily heat by evaporative cooling, would be stimulated to greater and still greater effort, no argument concerning the discomfort of the victim is necessary.

It may be possible that such a situation affects the senses partly, at least, because of an abnormal electrical predicament. There seems to be some indication that the discomforts of an antipodal situation; i.e., an extremely low humidity condition, resulting in excessive evaporation from the skin, the eyes and the mucous membranes is also due to electrical predicament, though the phenomena would be in reverse order.

It has been demonstrated by the chemist that the so-called *ionization* identified with the intensity of acid activity produces profound changes in certain protoplasmic colloids. Apparently, no one knows exactly what these changes are, but it has been proved that the same results may be obtained in other ways than through alteration and control of the potential activity of an acid solution. In other words, though *ionization* is indubitably a concomitant of *acid activity,* it is perfectly possible that the phenomenon is not restricted to that source—that it may, in fact, be a product common to all activity.

Is Electricity of Paramount Importance in Air Conditioning?

When heat is transformed from a sensible or a latent state a tremendous amount of kinetic energy is evolved or involved, as the case may be. This predicates activity—a great deal when liquid water changes to vapor, and vice versa; less when solidification and melting are concerned. But potential energy is the same in either case, and is equivalent to the amount of heat concerned in the transaction. And since all research in the field of physics leads toward the conclusion that heat and energy and that mysterious agent called *electricity* form a sort of trinity, in which three are one and one is three, it does not seem impossible that electrical predicament may actually be the paramount consideration in any form of air conditioning endeavor.

Comfort in Various Air Environments

It will not be disputed that data upon which the engineer might base his endeavor toward perfection in atmospheric environment are largely of empirical origin. Empiricism has both virtues and defects. Very often it makes the mistake of attributing an effect to the wrong cause. It would require not an engineer but a psychologist to produce the same degree of comfort for all human beings with an identical *effective*—so called—temperature, secured in one instance with 62 deg. dry bulb and 100 per cent relative humidity and in another by 70 deg. dry bulb and 26 per cent relative humidity, the air movement being the same in both cases.

There are two reasons why the lower temperature condition would fail to produce the same degree of comfort as the higher. It would require approximately 7 per cent more heat to warm a given volume of air 1 deg. at 60 deg., saturated, than it would at 70 deg., 26 per cent saturated. And since the temperature head between the body and the surrounding air would be increased by ten degrees, though the total heat of the mixture of air and water vapor at the lower temperature level would be some 31 per cent the greater, the *sensible* heat at 70 deg., 26 per cent saturation would be about 6 per cent higher than that present in air at 62 degrees, 100 per cent saturation.

As was before mentioned, the human body is not *hygroscopic,* using the botanist's term. While it is capable of furnishing liquid water for vaporization, it does so largely because of the activity of its sweat glands. These are certainly not very active at a temperature 36 degrees below that of the blood.

It is therefore idle to expect the excess of sensible heat in the 70 degree, 26 per cent saturated air to become latent when there is little or no liquid water to be evaporated from the skin.

Nor can we expect any transition of latent to sensible heat in the 100 per cent saturated air via the condensation route. The body will not absorb latent heat, since it cannot absorb the liquid water concomitant to the transition.

Certainly, if the temperature of the body were below that of the air, moisture might be condensed on the skin and an equivalent quantity of sensible heat recovered, as often happens with a pipe carrying a cold fluid, or with a glass of ice water set in a warm room. But it is well known that to heat the animal body is never the task of the engineer. He may add heat to the air, thus lessening the loss of bodily heat, and he should by all means give adequate attention to the nature of this heat and the efficacy of its application.

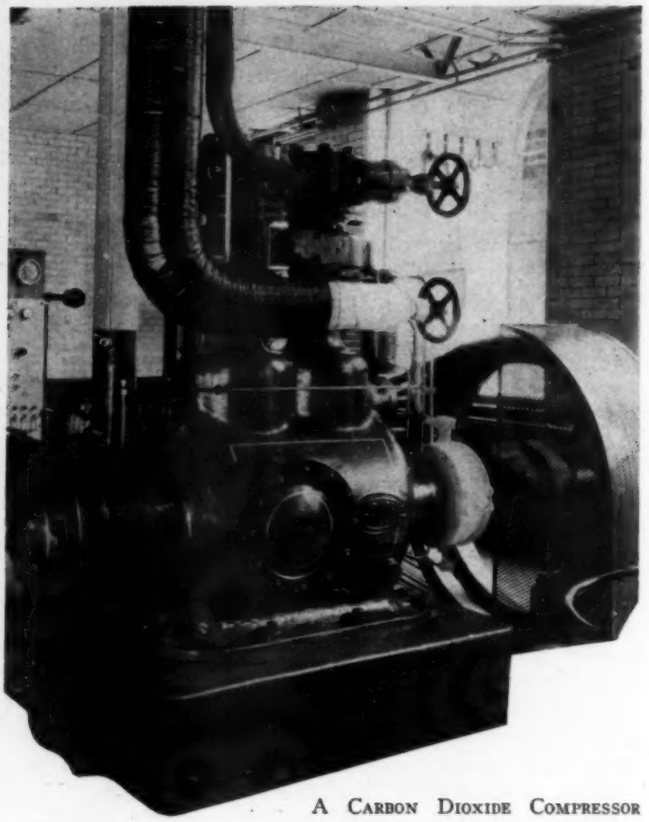
Unquestionably, in an environment carrying a high ratio of sensible to latent heat liquid water is rapidly evaporated from the lungs, the lining of the nose and throat, and the eyes, with a consequent transition locally. Local cooling is certainly high, though possibly compensated for by an influx of warm blood to the affected parts.

Engineer Must Know Causes

It is not for one unfamiliar with physiology and anatomy to denominate the nature of the phenomena, but anyone who has ever experienced it knows that the results are not conducive to comfort and tranquillity. Perhaps the hot, damp environment of the unconditioned auditorium is even worse, and perhaps in either case an unfavorable electrical predicament plays a rather important part.

Perhaps it is the job of the engineer to provide the conditions specified. But the average man who can decorate a contract and a check in a manner satisfactory to everyone concerned knows little and cares less about "conditions" as indicated by thermometers, anemometers and hygrometers. The "conditions" that interest him are those that produce the maximum of comfort and efficiency, positively, dependably and economically. He is interested in effects, not causes; while it is the job of the engineer to be interested in both.

Selecting CO_2 Air Cooling Equipment



A Carbon Dioxide Compressor

By A. N. Chandler*

SUCCESSFUL and uninterrupted summer operation of theaters, restaurants, banks, department stores, etc., makes the mechanical air cooling plant a necessity. Therefore, the selection of the refrigerating or cooling equipment for each installation is of great importance. Mechanical air cooling equipment is established as a necessity. It is no longer to be considered prohibitive or an unnecessary expense which is to be cut out of a budget at will. It is one of the money-makers to every industry which can use it. Recent installations show the progress in refrigeration, while great improvement has also been made in temperature and humidity control equipment. In this article, we are interested only in the proper selection of the CO_2 refrigerating equipment necessary for a typical air cooling job.

Let us select the refrigerating equipment for a theater job of 184 tons:

* Frick Co., Waynesboro, Pa.

Compressors

Perhaps the heaviest, bulkiest piece of apparatus in each installation is the compressor unit. This usually consists of either a vertical single acting, vertical double acting, horizontal reciprocating, or centrifugal compressor unit.

It is usual practice where a large installation is made to install the refrigerating units in duplicate. This would mean, in this case, two 90-ton units, each arranged to operate as a single unit if necessary. This makes the operation simple and at times allows a large saving in power and operating costs.

Locating the Compressors

Having two units, we must find sufficient floor space for the compressors, motors, if synchronous motor-driven motor generator sets are used, and, in any case, electric starting panels. We must locate the compressors where the noise from them cannot be transmitted to the building steel work, or to piping in the building. This means

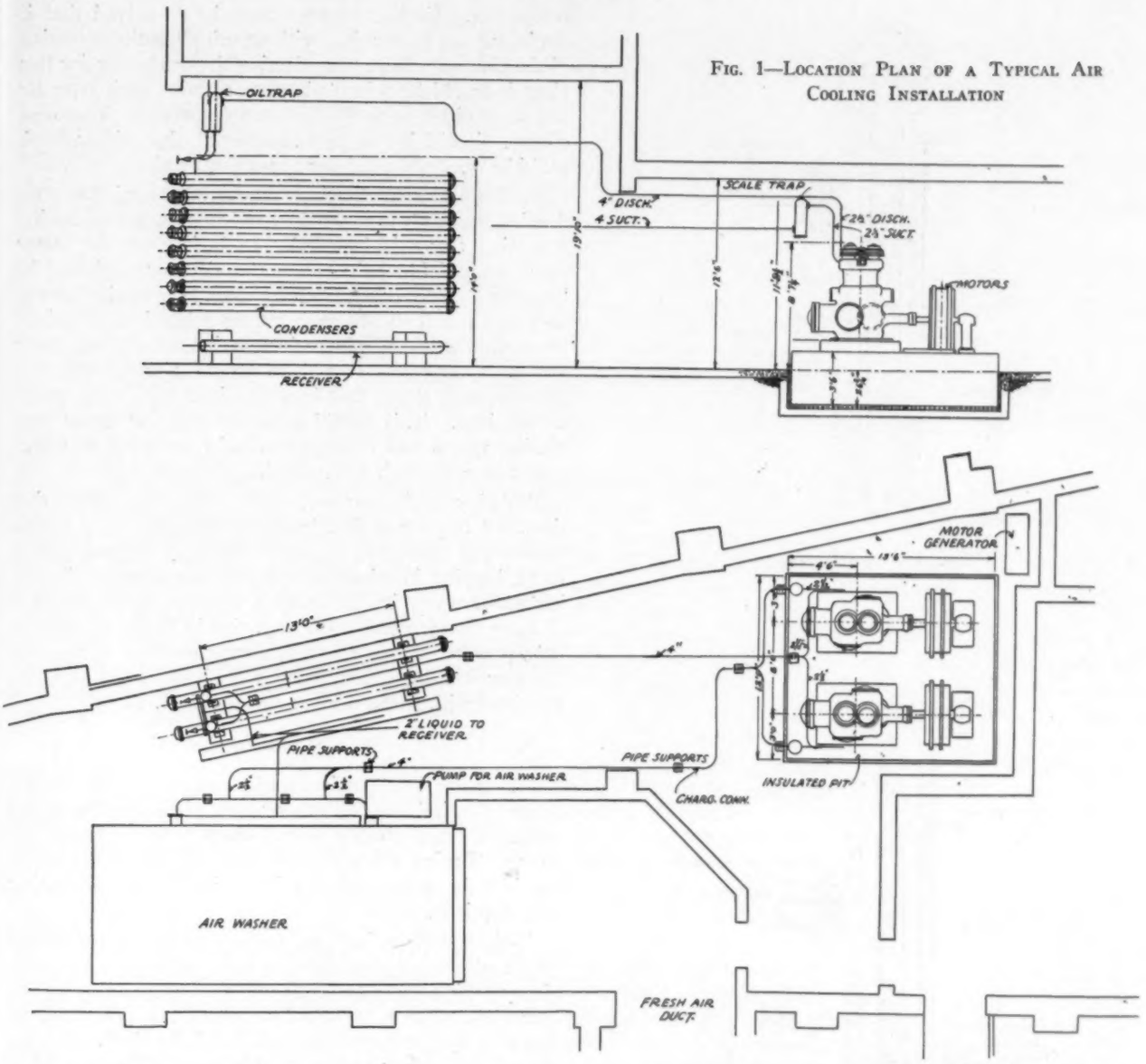


FIG. 1—LOCATION PLAN OF A TYPICAL AIR COOLING INSTALLATION

the foundations must be built so they can be insulated thoroughly. If located near columns, the foundations must also be carefully insulated from them. The compressor units must be so located that they are readily accessible for the operation of control valves. Furthermore, they must have space for pulling pistons and for making such repairs as are necessary on equipment of this type. It is preferable to have the starting equipment for the motors located as near the units they control as possible, the ideal situation being such that the engineer can control his plant with the greatest ease. Fig. 1 shows a typical layout for two 90-ton units. The foundations were built in an insulated pit, lined with cork.

Condensers

Condensers of either the double pipe or shell and tube type are to be located as near the compressor units as possible. This not only shortens the pipe lines, but makes the control accessible. Condensers should be mounted rigidly or have sufficient foundations or piers to hold them from any tendency to vibrate. They must also be located so that there is sufficient room in which to remove the pipes or tubes which in time will become

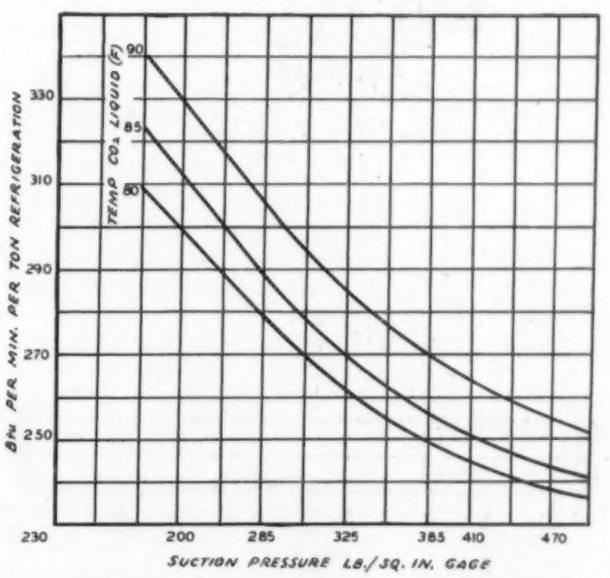


FIG. 2— BTUS REMOVED IN CONDENSER PER TON REFRIGERATION—SINGLE COMPRESSION

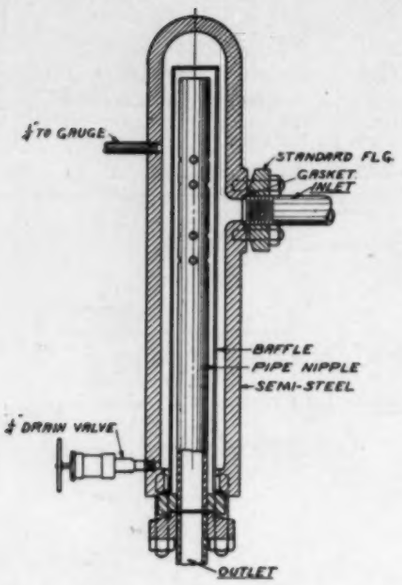


FIG. 3—CO_2 OIL TRAP

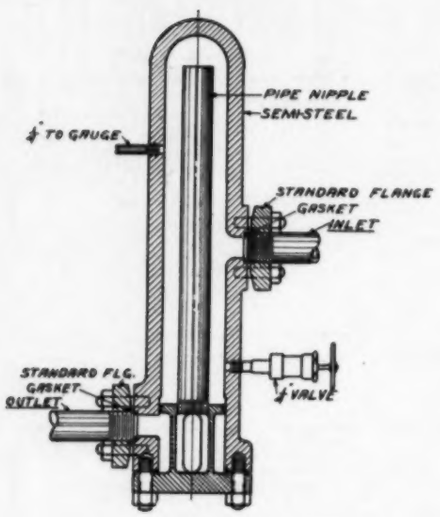


FIG. 4—CO_2 SCALE TRAP

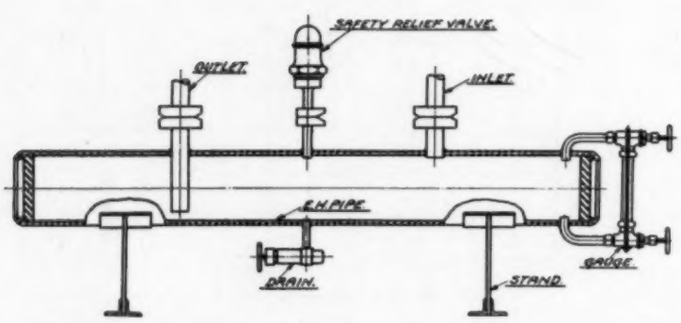


FIG. 5—CO_2 RECEIVER

rusted out. Each condenser must be so valved that it can be cut out at any time without crippling the operation of the plant as a unit. In selecting the condenser for this plant it would be wise to have two units, each separate and each piped to a 90-ton compressor unit. These, of course, should be so cross-connected that either bank could be operated on either compressor unit.

In the majority of plants the water used for condensing purposes is expensive, and as a means of saving a cooling tower is installed. In laying out the water lines, an individual line from the city mains should be brought to the condenser, in case the cooling tower or water pump should ever get out of order. Furthermore, this would eliminate installation of a spare water pump. Condensers should be provided with thermometers on each stand, that is on the inlet and outlet water connections. This would naturally call for water regulating valves and headers to obtain the same temperature range through each stand.

Probably condensers are, next to the compressor units, the most important link in the cooling plant. For this reason the condenser capacity should be figured amply large, keeping in mind the extreme temperature of cooling tower water in the summer season. It always pays to have excess condensing capacity in order to cut the horsepower required to drive the compressor during the summer season. Each condenser installation must be provided with an oil drain and a water drain.

Size of Condenser

The size of the condensers in square feet of surface depends entirely on the temperature of condensing water available from cooling tower, deep well or from city mains. Taking the plant we have of 184 T.R., let us consider the steps necessary for designing the condensing unit required.

Assume that we have an average condensing water of 75 F and that a spray tower cooling water system is used. Every cubic foot of gas delivered to the condenser by the compressor has a certain heat content which must be removed by the condenser water and liquefied. For example, consider a carbon dioxide plant which has 85 F liquid from the condenser which can be obtained with 75 F water on the condenser. With this temperature liquid we have a condensing pressure of 1025 gage. From Fig. 2 we see that the condenser must remove approximately 245 Btu per ton or a total of 45,080 Btu. The following equation gives the amount of condensing water required:

$$\text{Amount of condensing water} = \frac{\text{Btu/T.R.} \times \text{total tons}}{\text{t.d.} \times \text{lb./gal.}} = \frac{245 \times 184}{10 \times 8\frac{1}{3}} = 540 \text{ gallons.}$$

(Note: t.d. equals temperature difference between inlet and outlet water temperatures.)

Using a heat transfer in the condenser of 300 Btu per square foot per degree per hour we can figure the condenser surface required as follows:

$$\text{Condenser surface} = \frac{245 \times 60 \times 184}{5 \times 300} = 1{,}830 \text{ sq. ft.}$$

Note: "5" equals mean temperature difference between inlet and outlet water temperature.

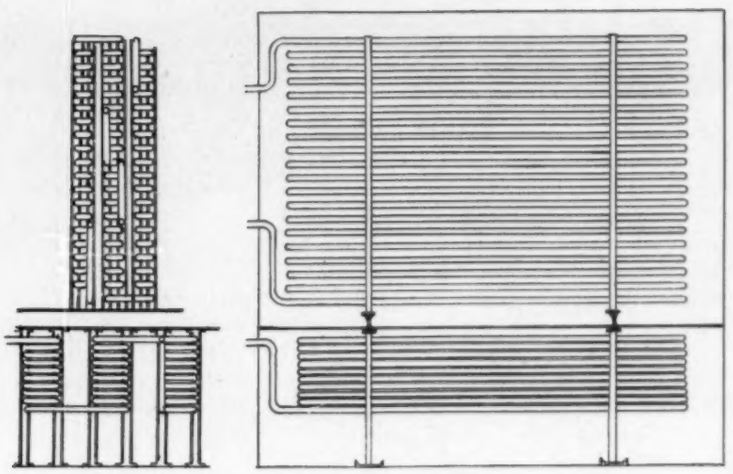


FIG. 6 — CO_2 EVAPORATOR UNIT

Oil Traps and Suction Separators

Every refrigerating installation calls for an oil separator in the discharge of each compressor unit. This separator is usually located as near the condenser as possible. The purpose of the oil separator is to catch any oil which may be taken over in the discharge gas from the compressor. It is provided with baffles which change the direction of the gas and upon which is deposited the oil. The oil drains off to the bottom of the traps and is taken out through a valve provided for this purpose. Fig. 3 shows a cross section of an oil separator used with a carbon dioxide plant.

The scale trap is inserted in the suction line from the evaporating units. Its function is to catch any dirt, scale, etc., which may come loose in the pipe during erection. It is provided with a removable screen. If this dirt were allowed to get to the compressor unit it would seriously damage the piston and valves. Fig. 4 illustrates a cross section of a scale trap in common use. Fig. 1 shows the proper location of the scale trap and oil separator in a typical plant.

Receiver

After the gas discharged from the compressor is condensed in the condenser it is collected in the receiver as a liquid for use in the evaporating units. The size of the receiver depends on the amount of liquid which is to be stored. In this particular plant a receiver eight inches in diameter by sixteen feet long would be used. It must be provided with inlet, outlet, *gage glass, and safety valve*. With the gage glass the operating engineer can at all times see the amount of charge in the plant and arrange the operation accordingly. Fig. 5 shows a cross-section of a CO_2 receiver.

Size of Receiver

The size of the CO_2 receiver can be determined by estimating the maximum amount of CO_2 gas that is held by the largest evaporating unit. It requires 2.3 ft. of 1¼-in. pipe for one square foot of external surface or 96.275 ft. of pipe to contain one cubic foot.

The size of the receiver selected, 8 in. diameter by 16 ft. long, has a volume of five cu. ft. When filled with 85 F CO_2 liquid it can hold 280 pounds of CO_2. The receiver would then be capable of holding the gas pumped from approximately 480 ft. of 1¼-in. pipe. As previously mentioned all the pipe in the washer is divided into small units, valved so that it can readily be pumped out. In cases of emergency CO_2 can also be stored in the condenser.

Evaporating Equipment

The air washer, being the unit in which the refrigeration is applied, ranks equally in importance with the compressor. Its duty is to transmit the refrigerating effect produced by the refrigerating machines to the air. Air washers, as used today, are of two general types, direct expansion and those using water cooled by separate equipment. The installation for which we are selecting equipment was of the former type, and will be described in more detail.

In the so-called direct expansion system of air cooling the cooling or evaporating unit is installed directly inside the air washer, in the air flow. The temperature of the gas inside the coils being lower than the temperature of the air, a transfer of heat takes place from the coil to the air. This in itself would cool the air, but frost and ice would eventually form on the coils, which would cut down the heat transfer. To overcome this it is necessary to flood the coils continually by means of water which is distributed over them by troughs. It is also customary to install banks of spray nozzles in front of the expansion coils. This helps reduce the refrigerating load on the cooling coils, and makes it possible to shorten the length of the coil chamber.

Air washers of the direct expansion type may be designed for the use of 1¼-in. direct expansion pipes, the amount of which, per ton refrigeration, depends upon the air conditions desired. The average installation uses approximately 35 lineal feet for each ton of refrigeration required. Fig. 6 shows a typical coil designed for air washer use. The pipes are spaced on from 4½- to 5-in. vertical centers. They are placed across the cooler with all connections on the outside of the cooler. A trough is usually placed above each section of coil, which can be made adjustable and can be arranged to give a uniform stream of water over the entire length of the pipe.

Usual practice is to allow from 1½ to 2 g.p.m. of water flow per ft. length of coil. Direct expansion coils in washers are placed on from 8- to 15-in. horizontal

centers, depending upon the space alloted in the washer and the amount of coil which must be used.

As previously mentioned, the air is drawn by the blower through the water spray and cooled and then forced out by the fan to the space to be cooled. The water cooled by contact with the coil falls to a tank where it feeds to the suction of a pump and is returned to the troughs and nozzles. The pump has a float control for letting in make-up water. Thermometers are installed in the water lines to determine the proper operation of the plant, and the proper suction temperature to carry.

Cooling Tower and Water Pumps

The installation of a cooling tower for saving water is almost a necessity for most air cooling plants.

For efficient operation it usually requires approximately 2 sq. ft. of horizontal active tower area to handle each ton of refrigeration. For the case of a 184-ton plant, then, it would require 372 sq. ft. of active tower area. A tower 12 ft. wide would be 31 ft. long at the base. The actual floor space required might be 7 ft. longer and wider due to the tower construction.

Cooling towers are generally guaranteed, when operating at a capacity of approximately 1½ gal. of water per min. per sq. ft. of area, to effect different temperature reductions depending on wind velocity, water temperature on and off tower, and wet bulb temperatures. For example we will consider this case wherein water goes to the tower at 85 F. From data prepared by manufacturers we could expect the water to leave the tower at a temperature of 74 F.

The capacity of the condensing water pump has already been determined in the paragraph under "Condensers." It should be located in the engine room at a point where it will be readily accessible. The return suction line should have a screen installed to eliminate foreign matter from clogging the impeller. The pump should also be installed on insulation mats to prevent noise.

The Pipe Connections

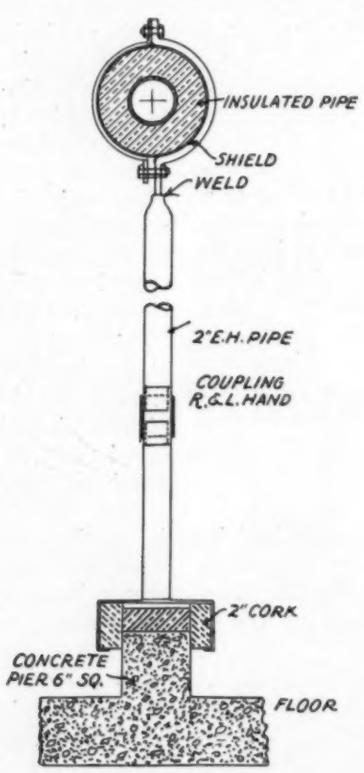
A plant consisting of several units naturally requires pipe connections designed not only to meet the operating pressures used, but also to meet various pressure codes and municipal regulations. To eliminate as many joints as possible this particular job was welded. Where high pressures are used special high test valves and fittings must be utilized. The valves must be so designed to be free from leaks through the stuffing box and be constructed with a gland that can be readily tightened.

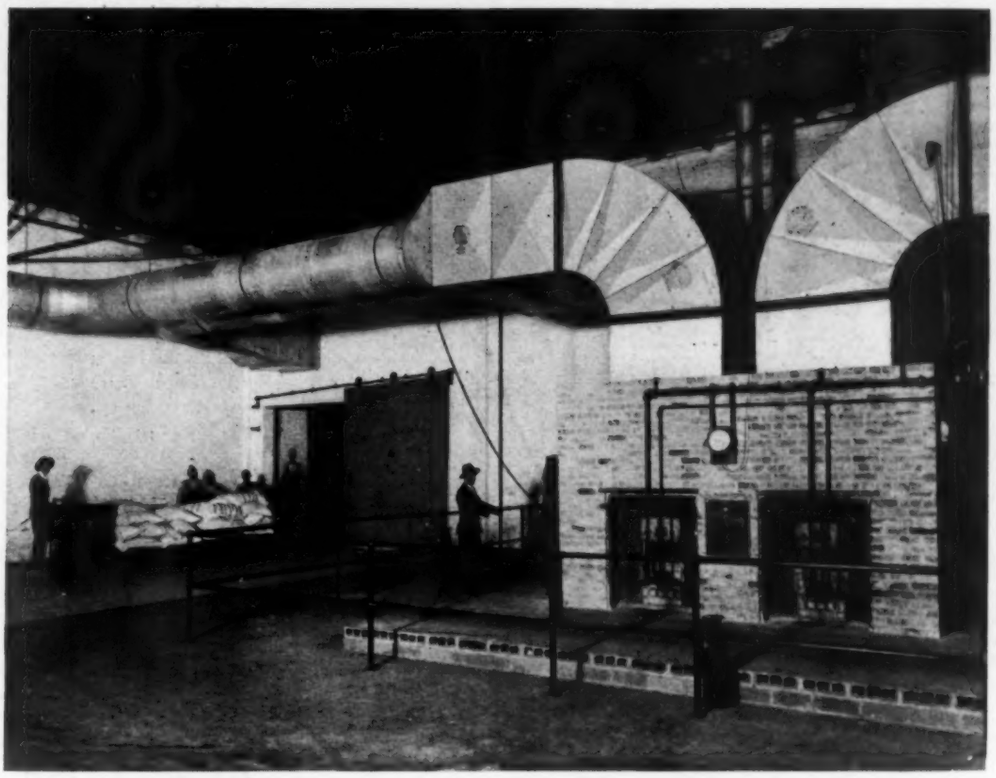
After each installation has been erected and thoroughly tested for tightness it is necessary to insulate the suction lines, that is, those carrying the low temperature gases from the evaporator. These gases vary in temperature from 25 to 50 F. The proper insulation should be applied in the best manner possible and be made watertight.

Conclusion

As cooling equipment is used only for 125 to 150 days per year approximately, it is highly important that it function 100 per cent each of these days, and to do this it must be carefully designed, properly installed and operated efficiently.

FIG. 7—SUPPORT FOR CO_2 PIPE LINE



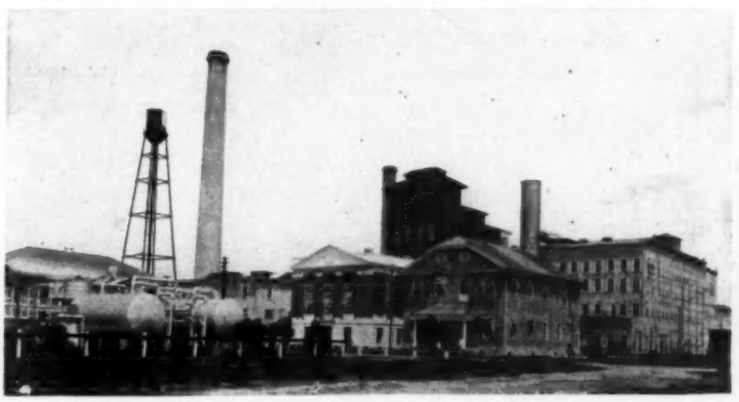


THE HEATING PLANT CONSISTS OF TWO 44-IN. DIAMETER BRICK SET FURNACES

Warm Air Heating Plant Protects Sugar in Storage

COLONIAL SUGARS COMPANY introduced a new manufacturing idea to the sugar industry early in 1929 when it installed in its refinery at Gramercy, La., a modern warm air heating plant. It is thought to be the first time in the history of the industry that artificial heating has been used to control humidity and temperature in a sugar refinery. The job was undertaken to solve an age-old refinery problem.

The purpose of this installation is to avoid the absorption of moisture (and subsequent caking and hardening) in refined sugar in storage. Granulated sugar is hygroscopic and when exposed to moist atmospheric conditions, such as prevail at the refinery, particularly in winter, it absorbs moisture, forming a very slight film of syrup on the crystals. Any sharp drop in temperature will cause this syrup film to crystallize, cementing the grains of sugar together into a hard cake. The warehouse heating system serves a double purpose. When the inside temperature is higher than the outside the relative humidity in the warehouse is lower and there is

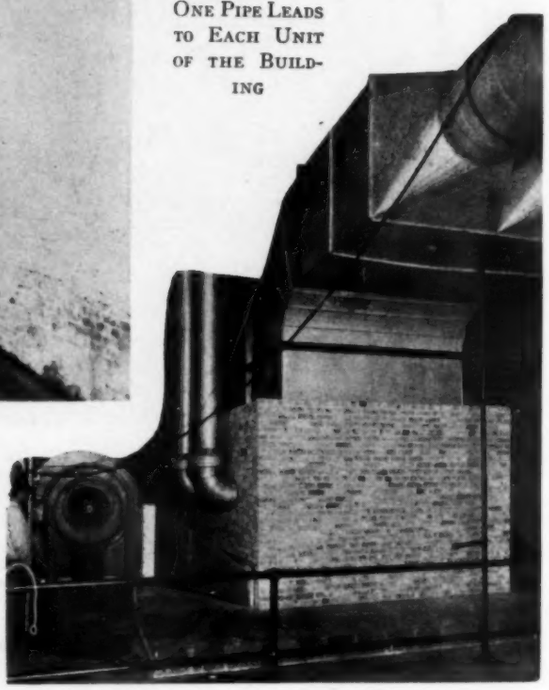


THE GRAMERCY REFINERY OF THE COLONIAL SUGARS COMPANY

OPENINGS ARE ARRANGED AT PROPER INTERVALS TO INSURE AN EVEN DISTRIBUTION OF AIR

The plant heats two warehouses, 100 ft. wide x 200 ft. long x 20 ft. high. The capacity of each warehouse is 140,000 bags of refined sugar.

ONE PIPE LEADS TO EACH UNIT OF THE BUILDING



little or no tendency for the sugar to absorb moisture.

Furthermore, by keeping the sugar warm at all times, caking cannot possibly occur. The system has served its purpose admirably. It has been found that no exact humidity control is necessary. In the winter time the temperature in the warehouse is kept above 45 F and preferably around 60 F; in summer the warehouse is warmed at night to a few degrees above atmosphere, while in the day time the fire is cut off and the warehouse doors opened (on dry days only).

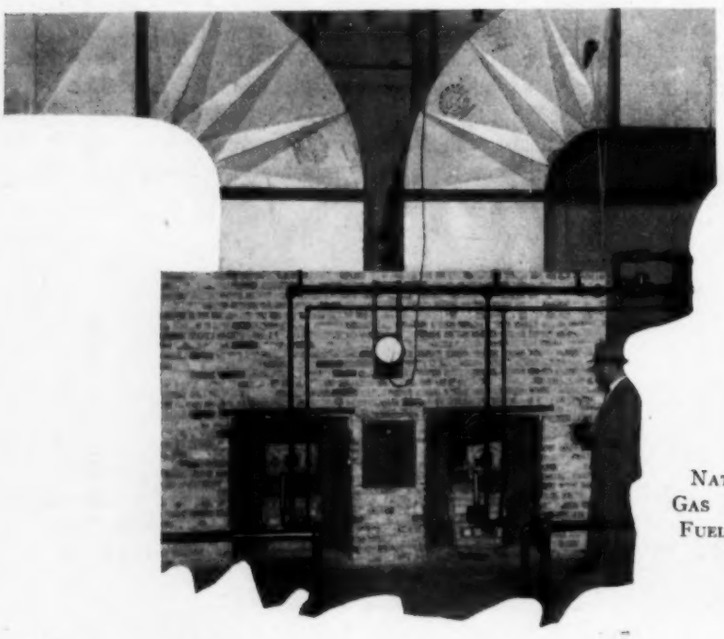
The installation has entirely avoided all necessity for reconditioning or reworking sugar, no matter how long it may have been in storage.

The heating plant consists of two 44-in. diameter brick set furnaces, placed between the two warehouses. They burn natural gas; about 3,066 cubic feet per hour in extreme conditions. A galvanized pipe leads from the plant into the warehouses, there being one pipe for each unit of the building. It runs down through the center of the building, with openings arranged at proper intervals to insure an even distribution of the air.

A single inlet, single width blower in each pipe forces the warmed air through the distribution channel; another fan pulls fresh air into the furnace for the proper tempering process.

Because this is the first refinery to use such elaborate means of controlling warehouse temperature, the system is being watched with much interest by the sugar industry.

The installation was made by the Blattman-Weeser Sheet Metal Works, Inc., of New Orleans, La., and the engineer was Fritz Gutknecht of that organization.



NATURAL GAS IS THE FUEL USED

Flexibility Keynote of Air Conditioning System

for Combined

Office Building and Laboratory

MOTOR CONTROL EQUIPMENT FOR THE FAN WHICH VENTILATES THE BASEMENT

By W. R. Woodward*

VARYING amounts of heat generated in the electrical testing and research laboratories and a location in an industrial valley having an unusually dirty atmosphere combined to make the air conditioning of the new Westinghouse office building and laboratory a problem which was studied with care. The structure is located at the East Pittsburgh works of the Westinghouse Electric & Manufacturing Company near other large industrial plants and steel mills; general studies made in the district have indicated that an average of over 1,000 tons of dirt—ash, carbon, and tar—are deposited annually per square mile; in the industrial valley, the rate is undoubtedly higher.

In addition to containing general and private offices, this new 12-story building also houses electrical laboratories, necessitating considerable flexibility in the air conditioning system. The heat load varies from day to day, as may the arrangement of partitions. Control to each unit area is a feature of the job.

Careful Studies Made

A preliminary study was made, in order to determine the expected benefits of air conditioning, and it was found that in the unconditioned offices the air during the heating season was extremely dry. As a result, it was necessary to maintain a high temperature—about 76 F—for comfort, resulting in a waste of heat and promoting colds and other disorders on the part of the occupants[1].

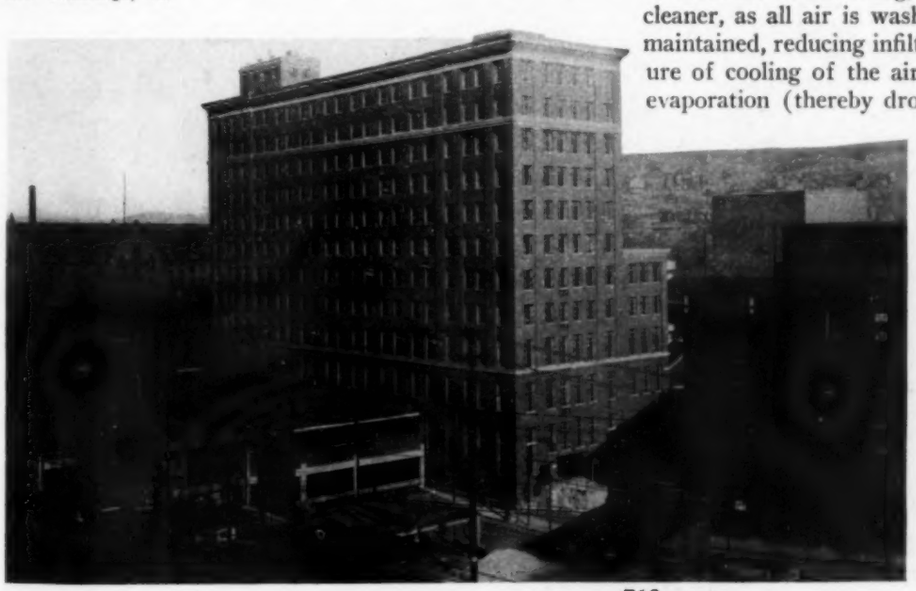
As no automatic regulation was provided in the unconditioned offices, temperatures often ran as high as 80 F due to persons seated near windows and doors insisting on more heat. In summer time, windows were opened, allowing large quantities of unhealthful and annoying dirt to enter, and permitting gusts of wind to scatter papers, etc.

Advantages of Air Conditioning

The decision to air condition the building was based on the advantages of controlling the humidity accurately in the winter, keeping it at 35 per cent or more and maintaining a temperature of 73 F, which was thought proper for "average" comfort. Also, with locked windows and proper arrangement of control apparatus, the right adjustments could be made by authorized parties only.

With air conditioning, air inside the building is much cleaner, as all air is washed and an internal pressure is maintained, reducing infiltration through cracks. A measure of cooling of the air is obtained in the summer by evaporation (thereby dropping to the outside wet bulb) in addition to the cooling effect of proper air motion. Provisions are made for the future installation of refrigeration equipment.

After due consideration the advantages of air conditioning were deemed sufficient to justify the necessary



THE NEW WESTINGHOUSE OFFICE BUILDING AND LABORATORY

* General engineer, Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa.

[1] Heat is wasted at the higher room temperatures by increased transmission losses through wall and stratification of room air. The total heat in Btu per lb. of air remains essentially the same when the temperature is lowered and the humidity increased. (See psychrometric chart.)

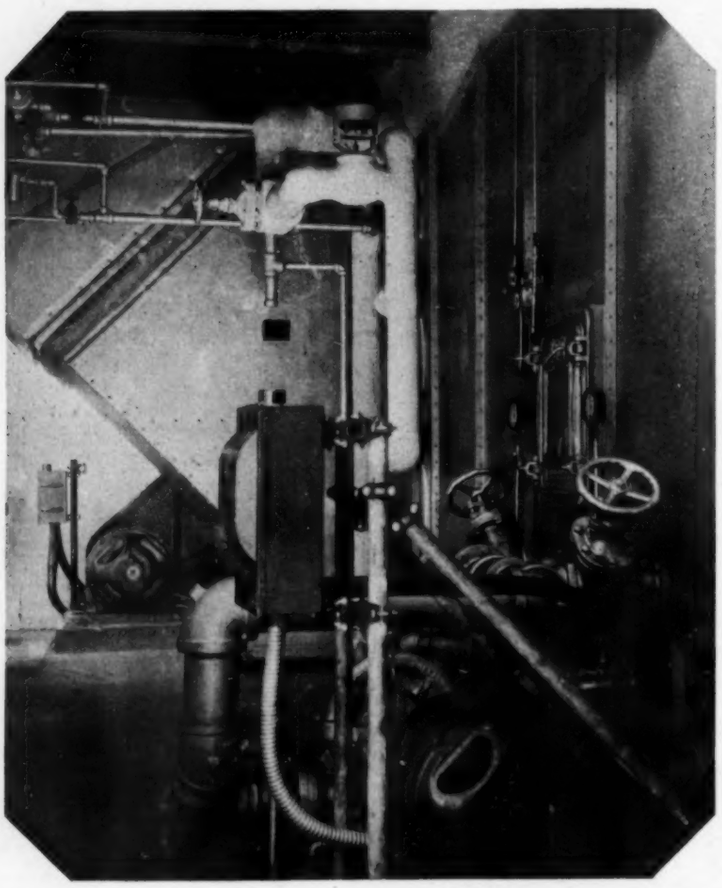
capital investment and operating charges, as it was felt that increased efficiency resulting from these improved conditions would ultimately more than balance the cost of the system.

Project Treated as Two Separate Problems

The project was treated as two separate problems, one applying to the lower six floors utilized for laboratory purposes and the other applying to the upper six floors utilized entirely for office purposes.

Air Conditioning the Laboratory

The areas devoted to laboratory facilities will house a great variety of activities. In general, the laboratory spaces are devoted to development of electrical equipment for commercial production and for solving such electrical and mechanical problems as arise during development stages. This means that the work will vary from day to day and from year to year, particularly from the standpoint of the amount of heat given off from the experimental tests. One day much electrical heat may be given off from resistors, motors, brakes and a multitude of other devices. The next day—or even the next hour—the test might be discontinued and the heat given off might be zero. Also, obnoxious gases, fumes or dust may arise in connection with certain testing, or electrical flashes or noises result, necessitating the enclosure of an area by partitions.



Air Conditioning Equipment for Office Floors, Showing Water Pump Motor and Control in Foreground and Fan Motor in Background

Unit Areas Individually Controlled

Due to these variable conditions it was felt that each unit area (in this case selected as a 20-ft. bay) must be made independent of its neighbor and individually controlled. This meant control of steam heat in winter time and control of amount of air introduced for cooling purposes in summer.

Each laboratory floor consists of eleven 20-ft. bays. A lengthwise corridor is provided on each floor (except the lower two floors) leaving a space 20 ft. wide on the narrow side and 48 ft. wide on the wide side. This unsymmetrical arrangement came about partly because of building features and partly to provide large unit spaces for certain activities.

Due to the large amount of heat which can be generated, particularly from electrical sources in such development work, the amount of air required for the laboratory areas is sufficiently large to justify a separate conditioning unit for each laboratory floor. The air conditioning unit is located in a room at the middle of the building on the narrow side and consists of intake and recirculating dampers, washer, fan, pump, temperature control and dewpoint control. Washed air (at about 60 F in winter) is delivered from the fan to a horizontal distributing duct over the top of the hallway. An outlet is provided into each 20-ft. bay on each side of the corridor. This outlet contains a volume damper and a heater unit. The damper and the heater valve are both controlled from a thermostat suitably located in the room or area served from this outlet. This will maintain the right temperature by shutting off excess air and introducing heat if the room is too cold and by shutting off heat and opening the volume damper if the room is too warm.

Air Discharged at High Level

Air from the outlets is discharged across the room at a high level so that in general drafts are avoided where they are objectionable. The heated air, on striking the windows is cooled, drops to the floor and passes back to the corridor through louvres in the bottom portion of the doors. From the corridor it is either drawn back to the fans for recirculation in the heating season or discharged through openings at the ends of the corridors during the cooling season.

Three Stacks Remove Heat

Three stacks about 36 in. square are provided for taking off obnoxious gases and concentrated heat from resistors, furnaces, and other equipment where heat can be carried off at high air temperature. This lessens the amount of air necessary to keep the room at reasonable temperature during the cooling season. In the winter time this heat can be saved by allowing it to mix with ventilating air.

The Question of Windows

Some consideration was given to the idea of constructing the building without windows. It is believed that artificial lighting can be arranged to give constant and desirable illumination; also, windows are harmful rather than beneficial from the air conditioning standpoint because they permit greater heat transfer than a wall. The idea, however, was abandoned as being too extreme at the present time for a twelve-story building. One laboratory floor, however, has no windows except at the ends on account of the necessities of building construction. Conditions on this floor will be studied carefully.

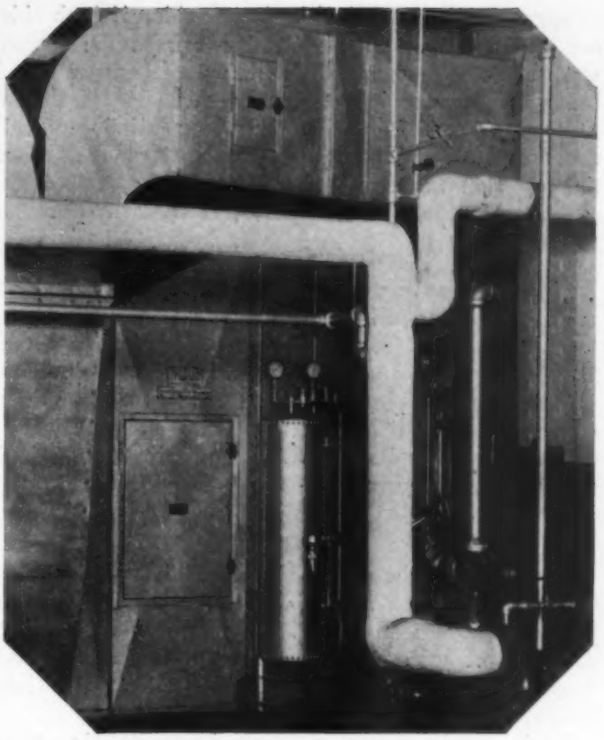
Control of Temperature and Humidity

The temperature control system embodies a pre-heater in the fresh air intake to keep the incoming air above freezing temperature to protect the sprays, a heater after the washer to maintain approximately 60 F in the air ducts and final heaters in each unit area, as mentioned above. By maintaining 60 F in the ducts a cooling effect can be obtained in any room where a large amount of heat is being generated.

The humidity control consists essentially in maintaining the temperature of the saturated air at a constant value. If this temperature is fixed and the air is heated to a definite room temperature the humidity is necessarily controlled (except for slight variations caused by external sources of moisture, such as process work or from people). Since the air and the water mix intimately in the spray chamber they quickly assume the same temperature. In the winter time, therefore, the water temperature (and hence the saturated air temperature) is controlled by the relative amount of outside (cold) air and inside (warm) air used. This proper mixture is readily controlled by automatic fresh air and recirculating air dampers. If the temperature tends to rise, more fresh (cold) air is drawn in. If the temperature tends to fall, more recirculated (warm) air is used and the water temperature responds to these changes.

To maintain humidity control in the summer time it is necessary to have a source of cold water, either from wells or by circulating the spray water through refrigerating equipment in order to control the water temperature (since no cold air is available). When cold water is used in the sprays the water temperature is often below the dewpoint of the fresh air and water in the air is, therefore, condensed. When this cold saturated air is heated by mixing with the warmer air in the room the average humidity has been lowered below that of fresh outside air.

No provision has been made in the present system for such cold water and the only cooling effect to be obtained in summer time will be the drop in temperature through evaporative cooling in saturating the fresh air.



THE WASHER FOR THE BASEMENT SYSTEM. THE LARGE PIPING IS THE RETURN TO THE VACUUM PUMP FOR THE ENTIRE BUILDING

OUTLET INTO LABORATORY ROOM SHOWING THE DAMPER MECHANISM, STEAM VALVE AND THE THERMOSTAT

Air Conditioning the Offices

The office portion of the building is approximately 50 ft. wide inside, leaving a space of approximately 22 feet on each side of the central duct or corridor.

The six office floors are used for general office purposes. The larger portion of each office floor is left as one large open room; some portions of each floor are partitioned off for private offices and conference rooms.

One air conditioning equipment serving the six office floors is located on the lowest floor. The other floors are supplied with air through two vertical flues, one for conditioned air from the equipment and the second for return air. The general nature of the equipment, the performance and control is the same as described above for the laboratories.

On each floor a central air duct is built along the

ceiling over the central aisle (or corridor if both sides are partitioned). Air is introduced from the fan air washer at the center of the building at about 60 F. A heater is provided in the duct at each side of the air supply so that each half of each floor is independently controlled from a thermostat suitably located at about the center of the area served. An air outlet from the center duct is provided every 10 feet on each side.

Control of Conditions

Since the entire area is used for general office purposes the sources of heat (lights, people and infiltration) are sufficiently uniform that one control point for a half-floor having an area of 4,000 sq. ft. is satisfactory. Some variation can be obtained by adjusting the amount of air admitted from the outlets. This adjustment, however, is a compromise only and cannot give definite individual control of small areas as is obtained in the laboratory areas.

Return Air

The air from the rooms returns to the exhaust shaft located in the center of the building through the corridor where the area is divided into private offices. Where there is no corridor the air returns to the exhaust shaft through the open room. No objectionable drafts have been encountered.

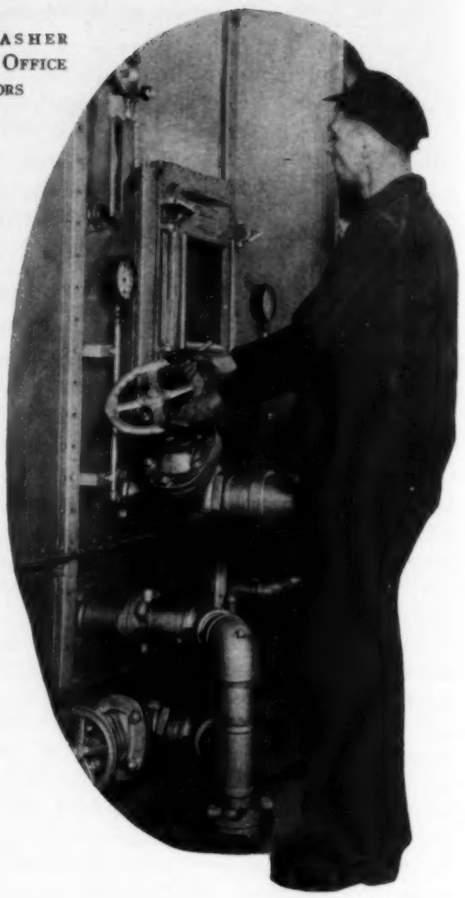
During the heating season a portion of the air returns to the washer and fan as required for humidity control. During the summer season all the air returns to the exhaust shaft and is exhausted at the top of the shaft to the outside of the building. The direction of air in the exhaust shaft is automatic, being controlled entirely by the automatic dampers controlling humidity.

Windows Padlocked

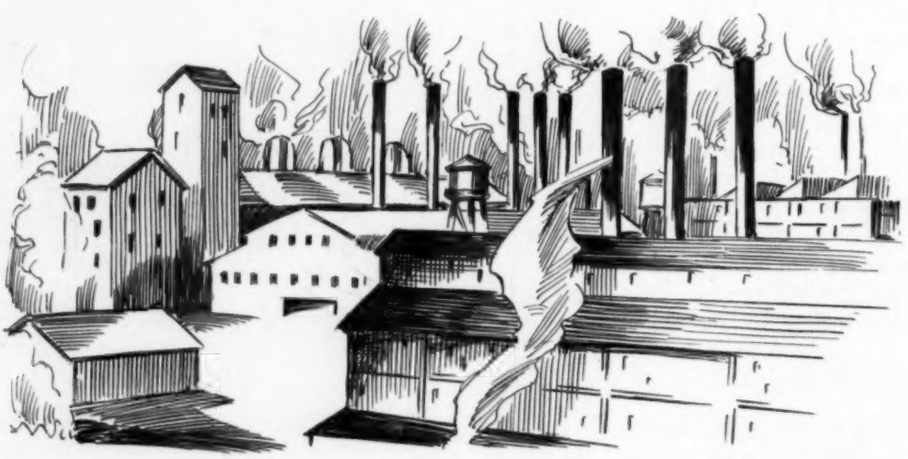
Satisfactory operation of the system during the heating season is dependent to a considerable extent on the ability to keep windows closed and to effect this each window opening is fastened by a small padlock.

Condensation on Windows

Experience through a portion of an exceptionally mild winter has shown little difficulty from window sweating provided the humidity is kept about 35 per cent. If the relative humidity is raised to 50 per cent the windows sweat continuously unless warmed by the sun or unless they are located on the shielded side of the building. The window sills are all made of terra cotta having a small trench so that what moisture collects on the windows is readily drained outside and does not drip onto the floor.



Air Washer for the Office Floors



Next month! A convenient and extremely practical method of keeping daily heating records for industrial plants, office buildings, etc. will be described. .....Arthur McCutchan's paper on engineering problems in the design of 1000 F steam piping will appear. .....Vern O. Knudsen's article on acoustics in air conditioning will be published.

"Open for Discussion"

What Size Reducing Valve?

SIZING of pressure reducing valves is a subject in need of attention from research organizations. The article, "Steam from Contented Reducing Valves,"[1] by Henry G. Schaefer, has brought this subject again to the front.

What Is Proper Size?

In the first place, what is really meant by the phrase, "Proper size of reducing valves?" Is it simply a valve amply large to do the work, or is it a valve of just sufficient size to do the work with the minimum amount of wire drawing and scoring? The article brought out the fact that valves larger than necessary for the load are undesirable and are more prone to give trouble than valves of lesser size so long as the capacity of the valve is not actually inadequate. It would seem that the users of such valves are fairly well agreed on this point. If this premise is accepted, it follows that the smaller the reducing valve is made, the more desirable it will be for heating loads where the full load seldom—if ever—actually occurs; that is, of course, within the limitations of being able to supply the full load should it ever be reached.

Disadvantage of Manual Control

In the above mentioned article, two reducing valves were suggested as a method of solving the difficulty or, at least, of applying a partial remedy to the trouble. The weakness of the suggestion lies in the manual control proposed, which is dependent on someone having the thought, time and disposition to change from one valve to the other every time there is a marked difference in the building demand for steam, whether this be from outside temperature changes, warming up in the morning, or an alteration in the hot water demand. It would seem that an improvement could be made in the proposed installation by arranging the valves to come in automatically, one after the other. This could be done by setting the reduced pressure on the smaller valve about ½ lb. higher than the reduced pressure on the larger valve.

Use of Thermostatic Valves

At the same time this would result in the pressure in the building being lower when the heavy load was on than it would be with a light load. The opposite condition would be much more desirable—so that idea is not ideal, although it sometimes is done. Another method would be to arrange thermostatic valves on the high pressure side of the reducing valves and to have these thermostatic valves controlled from thermostats set outside of the building in the manner shown in Fig. 1. It will be noted that the thermostatic valve set in front of the smaller reducing valve is also bypassed so as to permit this smaller valve being brought into operation again at very low outside temperatures. This is a rather complicated matter to make two reducing valves function when they should and not function when they shouldn't.

[1] Page 397, HEATING, PIPING AND AIR CONDITIONING, May, 1931.

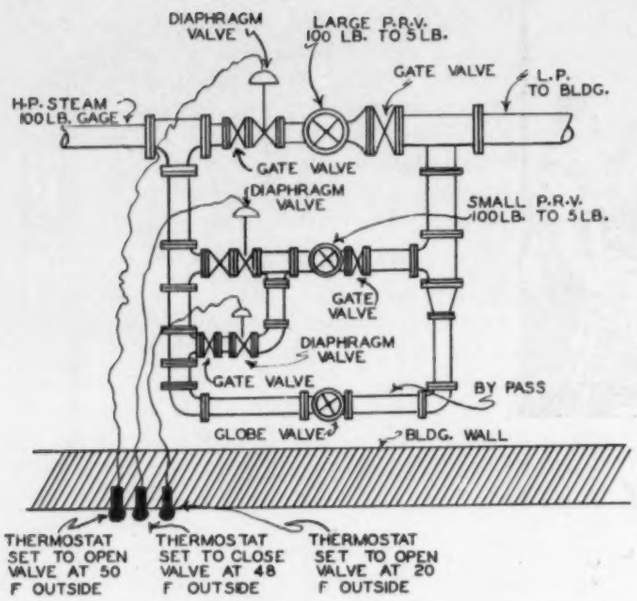

DIAPHRAGM VALVE
LARGE P.R.V. 100 LB. TO 5 LB.
GATE VALVE
L.P. TO BLDG.
H.P. STEAM 100 LB. GAGE
GATE VALVE
DIAPHRAGM VALVE
SMALL P.R.V. 100 LB. TO 5 LB.
GATE VALVE
DIAPHRAGM VALVE
GATE VALVE
BY PASS
GLOBE VALVE
BLDG. WALL
THERMOSTAT SET TO OPEN VALVE AT 50 F OUTSIDE
THERMOSTAT SET TO CLOSE VALVE AT 48 F OUTSIDE
THERMOSTAT SET TO OPEN VALVE AT 20 F OUTSIDE

FIG. 1—THERMOSTATIC VALVES CONTROLLING REDUCING VALVES

Maximum Velocity Through Orifice

For the sake of *simplicity* single reducing valve can not be beaten but it should be of such size as to just carry the maximum load and no more. The two valves proposed for a maximum delivery of 30,000 lb. of steam per hr. are in my opinion grossly oversized; in fact the 6-in. valve would carry the maximum load alone and there is a possibility that the 4-in. valve might get by; certainly a 5-in. would be ample. It is well known that gases discharged through an orifice reach a maximum velocity when the pressure on the low side of the orifice does not exceed about 58 per cent of the pressure on the high side. In other words, the discharging of 100 lb. steam through an orifice into a line at 58 lb. pressure would develop the maximum velocity obtainable through the orifice and a further reduction of pressure on the low pressure side would not increase this velocity no matter how low this pressure went, even if it dropped to a partial vacuum.

Formula for Lb. of Steam

Since heating systems seldom operate over 5 lb. pressures, it follows that any reducing valve delivering steam into the ordinary heating system may be sized on the maximum velocity basis provided the high pressure steam is at a pressure of 10 lb. or over. A great many conditions of this kind revolve around the reducing of steam from 100 lb. pressure to 2 to 5 lb. and, as the same velocity will hold down to 10 lb., the reduction from 100 lb. to 5 lb. or less has been considered in the following. The formula used is

$$W = \frac{A \times V \times 3600}{144 \times \text{cu. ft. per lb.}} \times E$$

Where

W is lb. of steam delivered per hr.
A is area of valve high pressure inlet and valve itself in sq. in.
V is maximum orifice velocity ft. per sec.
3600 is number of seconds in an hr.
144 is number of sq. in. in a sq. ft.
E is percentage of orifice efficiency.

In working out this formula two assumptions were made; the first is that the lift of the valve off the seat is sufficient to give an orifice area at least equal to the area of the inlet pipe size and, second, that the orifice efficiency is not less than 50 per cent. The 50 per cent is conservative and a larger size of high pressure line is assumed to be carried up to as close to the reducing valve as possible. Working this formula out for the various standard pipe sizes gives these values:

Capacities of pressure reducing valves—100 lb. to 50 lb. or less

Inlet Size of Valve "Nominal Pipe Diam." In.	Steam Supplied per Hr. Lb.	Radiation Supplied per Hr.—Equivalent Sq. Ft.
½	866	3,464
¾	1,576	6,304
1	2,459	9,836
1¼	4,263	17,052
1½	5,808	23,232
2	9,564	38,256
2½	13,623	54,492
3	21,041	84,164
4	36,285	145,140
5	56,971	227,884
6	82,336	329,334

But, will be asked, how do we know that these deliveries will be actually obtained in practice? Frankly, we don't! Nevertheless, these sizes have been used on a considerable number of jobs for a good many years and none of the jobs has fallen down from an inadequate steam supply.

It is possible, however, to do this much; the ratings obtained can be checked with the ratings which reputable manufacturers are putting on their valves and the results compared, although this may not be as enlightening as might be expected as there is little consistency exhibited in many cases.—*H. P. Adams.*

Author Comments on Velocity

MANUAL control of reducing valves, like that of perambulators, seems to me a necessity. With due respect for the ever-improving product of meticulously accurate manufacturing processes, and the fruitful efforts of our designing engineers in the direction of "foolproofness" of equipment, in the writer's opinion nothing but personal attention and responsibility will do in the operation of steam-handling regulating or reducing valves. Our national engineering progress may be attributed to fertile imagination, more than to any other reason. There is, however, a well defined limit beyond which ingenuity produces mechanical museums, rather than economical and practical plants.

In my original discussion of the subject, I stressed the "contentment" of the equipment. Quietness of the valve in operation is a necessity. The physical arrangement of piping has much to do with this phase of the matter, but it is a generally accepted axiom that velocities of steam in excess of ten thousand feet per minute meet with noisy protests of the gas. (I speak now of saturated steam at pressures above 10 lb.) That this is a fact has ample proof in past experiences. In the case of low pressure steam, especially in the case of heating risers, velocity is definitely limited to a maximum of 4800 feet. It follows that in the sizing of a reducing valve, the maximum allowable velocity through any part of the passage should not exceed 10,000 feet. It is on this basis that I have sized installations in the past with consistent success.—*Henry G. Schaefer.*

Heating·Piping
and Air Conditioning

American Society of Heating and Ventilating Engineers
JOURNAL SECTION

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A Study of the Combustible Nature of Solid Fuels

By R. V. Frost[1], Norristown, Pa.
MEMBER

Many consumers of fuel have experienced the trouble that accompanies the use of a fuel having a variable burning characteristic. "Variable burning" in the sense in which it is used in this paper, expresses a new thought in the study of the combustible nature of solid fuels. The phrase has not been employed by other research agencies in their reference to the subject but because of the different angle of approach outlined in this paper a different phraseology seemed essential for its proper expression.

WHILE the difficulty experienced with fuel having a variable burning characteristic is common to any fuel—solid, liquid or gaseous—the trouble has, from a practical standpoint, been virtually eliminated in the burning of liquid and gaseous fuels and coke. But in the use of natural coals, it remains one of the most serious and at the same time, the least understood of all the troubles with which the coal industry is burdened. It is a trouble common to both bituminous and anthracite coals and because of the increasing use of automatic burning apparatus, it is daily becoming of more vital consequence to the coal industry.

Many agencies are now endeavoring to obtain a solution of the problem. The U. S. Bureau of Mines has been engaged in a study of it for years, so far as it affects bituminous fuel, and every coal producer who boasts a research laboratory has delved into the subject. The statement has been made many times that coal will never "come into its own" until it is sold in liquid form, and many coal operators are today seriously considering the advisability of turning their properties into gas plants and transporting their product through pipe lines in the form of gas. Such is the importance of this subject in the eyes of those best informed.

Variability in the burning characteristics of coal makes itself evident in variation in the intensity of draft required to maintain a stated rate of combustion, and in the variability with which the fuel will ignite. The burning characteristics of coals are roughly classified as either hard or free burning, but there is no means at present in use by which a coal may accurately be classified as either hard or free burning. Appearance, while commonly used as such, is not an indicator. Neither are hardness, density, chemical analysis, location in the mine area or simple burning tests. Location is perhaps the best indicator, but this is not by any means positive, for frequently samples taken from the same lump are absolutely opposed in their burning characteristics. Sometimes the properties of a coal are so variable that its burning characteristics will change from one shovel to the next, and it is very difficult under the present methods of mining and preparation to obtain coal that is not more or less variable. In many cases, the variability is so extreme that of two samples from the same car compared on the basis of equal draft intensity, one will burn at a rate of combustion five or six times that of the other.

It is easy to imagine what a serious problem such variability in fuel becomes to manufacturers of boilers and automatic coal burning devices. This variability is undoubtedly the cause of many of the complaints raised regarding so-called inadequate boiler ratings and poor chimneys, in which case the boiler manufacturer becomes a party to the complaint. In the stoker industry the seriousness of variable burning characteristics is generally recognized and is regarded with genuine concern.

The correcting of the variability of burning characteristics is of more vital importance to the coal industry than is the production of a low ash coal, and of equal importance to uniform sizing, at least in the anthracite territory. In fact, a greater variability in sizing of anthracite is permissible than of variability in burning characteristics.

On account of combustion troubles encountered in the course of research in the operation of automatic coal burners, the writer and his associates began a study of this subject a year or more ago. Realizing its importance to the successful development of the stoker industry, the laboratory staff was led to conduct an investigation on the combustibility of all types of solid fuels. The efforts in this direction were rewarded with success in the development of a practical solution of the problem so far as anthracite and coke are concerned, but the method devised could not be applied with success to coking fuels.

In developing the method, an effort was made first to determine accurately how to classify the burning characteristics of a fuel and then to devise a method by which these characteristics could properly be indicated. Charts showing the relationship between draft and rate of combustion and the relationship between rate of horizontal ignition or fire travel and rate of combustion were devised. Several of these charts are included in this paper. (See Figs. 1 to 12, inclusive.)

Blending of Coals

The final step was to devise a system by which the practical use of the process could be made. In doing this, the laboratory determined the result of blending

[1]President, Frost Research Laboratory.

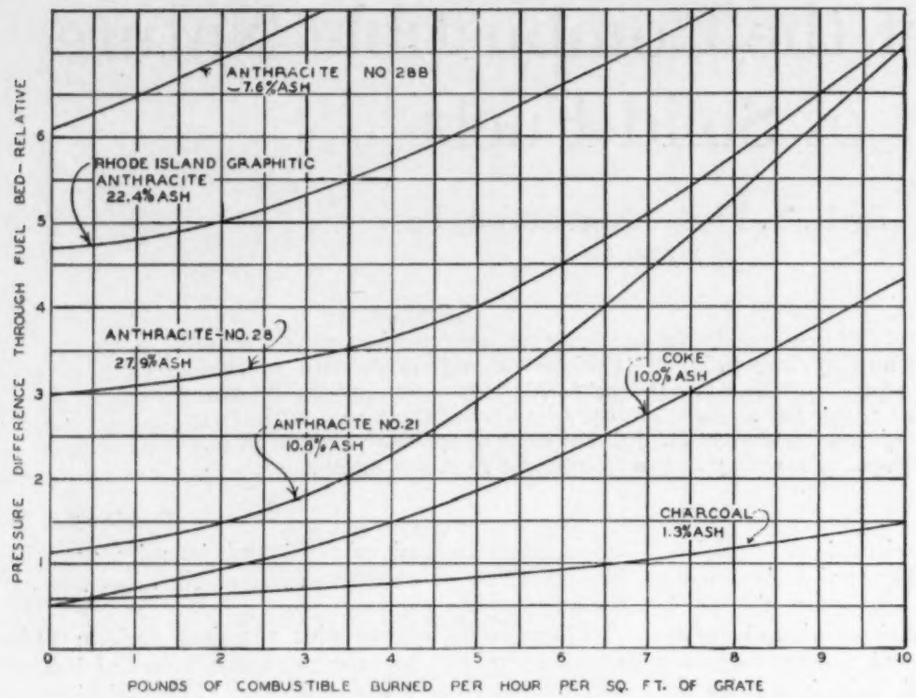

FIG. 1—RELATIONSHIP BETWEEN DRAFT AND RATE OF COMBUSTION FOR SAMPLES OF SEVERAL DIFFERENT FUELS

coals of different burning characteristics. Finding that blending overcame all the difficulties of variability, a method was developed by which the coal producer could so control his production that he could produce a composite coal so blended that its burning characteristics would continue with undisturbed uniformity from hour to hour, from day to day and from month to month, so long as he maintained an intelligent laboratory control of his production.

The first step in this research, which resulted in a laboratory system of testing, and the final step which is a matter of concern only to the coal producer, are omitted from further discussion in this paper. But the second step in the research, namely, the chart system showing the relationship of draft, rate of combustion and rate of ignition, together with the bearing of the entire problem upon the subject of heating, is a matter of interest to heating engineers.

Fig. 1 shows the relationship between draft and rate of combustion for samples of several different fuels. The lowest line on the chart represents a charcoal sample. This fuel required but little draft to maintain combustion and it is apparent that only a slight increase in draft was required to produce a high combustion rate.

Coke, which had a slightly higher resistance to high combustion rate, comes next on the scale. Then appear two anthracites, and then a sample of Rhode Island anthracite, a coal which offers so high a resistance to combustion that it is not offered for sale as a fuel. Yet this supposedly valueless anthracite apparently will burn much more readily than pure Pennsylvania anthracite, the

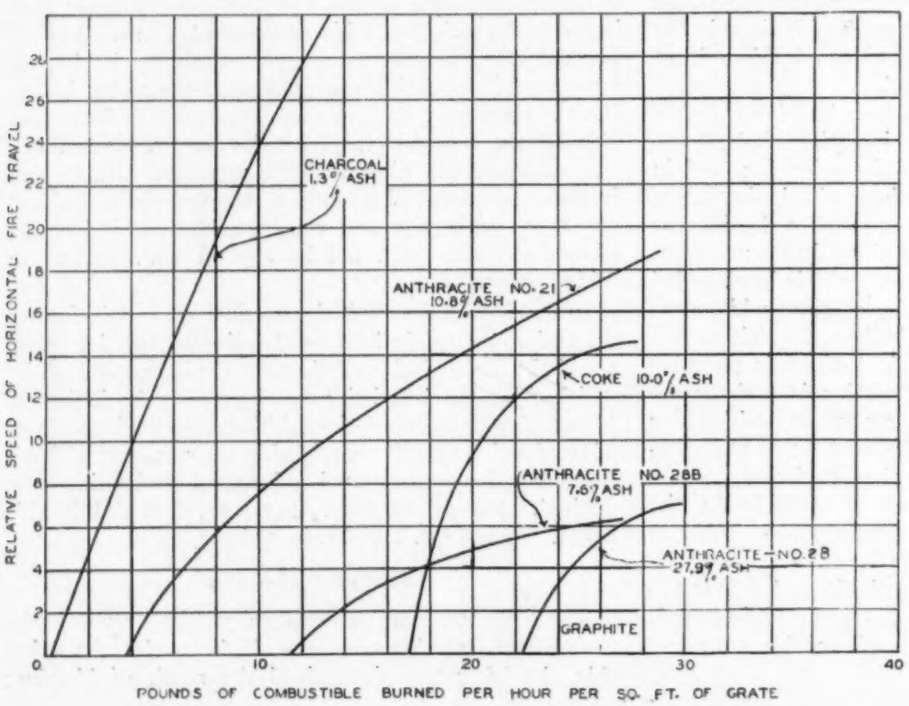

FIG. 2—RELATIONSHIP OF RATE OF HORIZONTAL FIRE TRAVEL AND RATE OF COMBUSTION

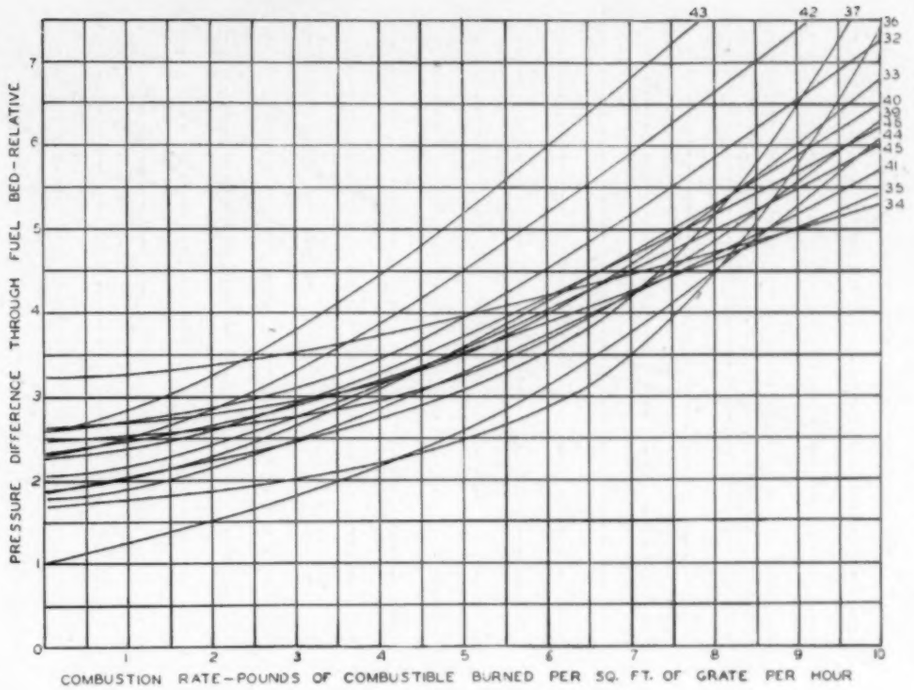
PRESSURE DIFFERENCE THROUGH FUEL BED - RELATIVE
COMBUSTION RATE—POUNDS OF COMBUSTIBLE BURNED PER SQ. FT. OF GRATE PER HOUR

FIG. 3—CURVES SHOWING RELATION BETWEEN RELATIVE DRAFT PRESSURES AND RATE OF COMBUSTION FOR SAMPLES TAKEN FROM EACH VEIN MINED AT *V* COLLIERY

curve of which appears at the extreme upper left corner of Fig. 1. This sample (No. 28*B*) is of unusual interest because it was a sample picked from a lump contained in sample No. 28, the curve of which is also shown. While the average ash of the entire No. 28 sample was 27.9 per cent as in the mine—this being a sample representing a certain location in a mine—sample No. 28*B* was selected from the whole as one representing a coal of excellent appearance and low ash content. Nevertheless, the burning characteristic of this fine appearing, low ash coal was inferior to the so-called valueless Rhode Island fuel.

Coal similar to sample No. 28*B* would require an exceptional draft to burn it, for by examination of the curves it is seen that sample No. 28, of which No. 28*B* is a part, will burn at rate of combustion of 9 lb per square foot of grate area per hour under a draft intensity which would not permit No. 28*B* to burn at the rate of even one pound per hour. It is also of interest to note that sample No. 21 is an anthracite that burns at low rates with as little draft as does coke and further that a coal of 28 per cent ash as represented by No. 28 burns almost as readily as a coal of 10 per cent ash as represented by No. 21.

Horizontal Ignition

Fig. 2 shows the relationship between rate of horizontal fire travel and rate of combustion. The *rate of horizontal fire travel* as used in this paper is an index of the rapidity with which the fire will spread from one part of the fuel bed to another. It is useful to indicate the rate of ignition and a coal of quick ignition is of course a free-burning coal.

In Fig. 2, charcoal again appears superior but coke is

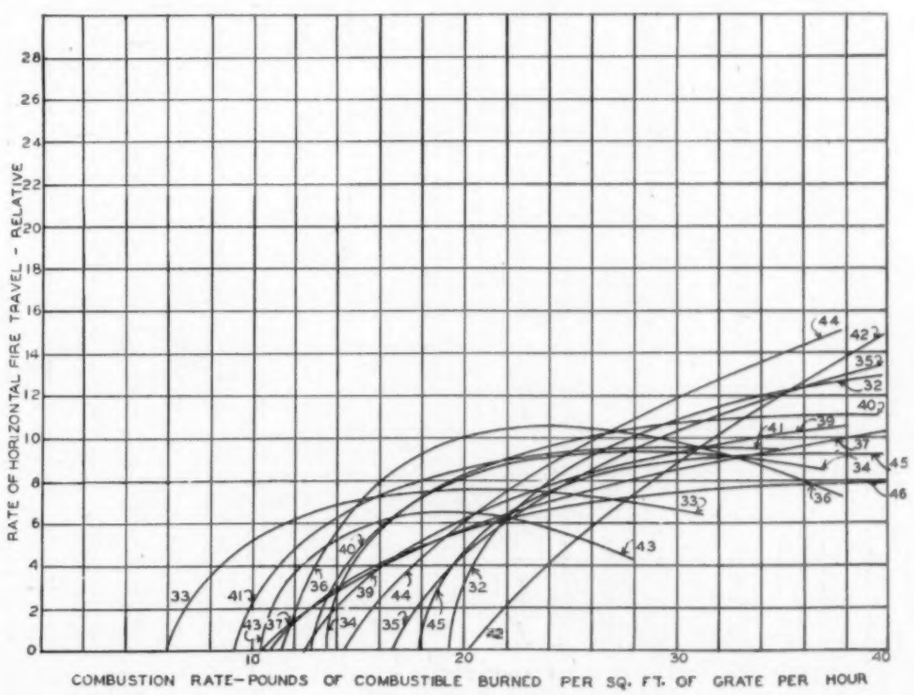

FIG. 4—CURVES SHOWING RELATION BETWEEN RELATIVE RATES OF HORIZONTAL FIRE TRAVEL AND RATES OF COMBUSTION FOR SAMPLES TAKEN FROM EACH VEIN MINED AT *V* COLLIERY

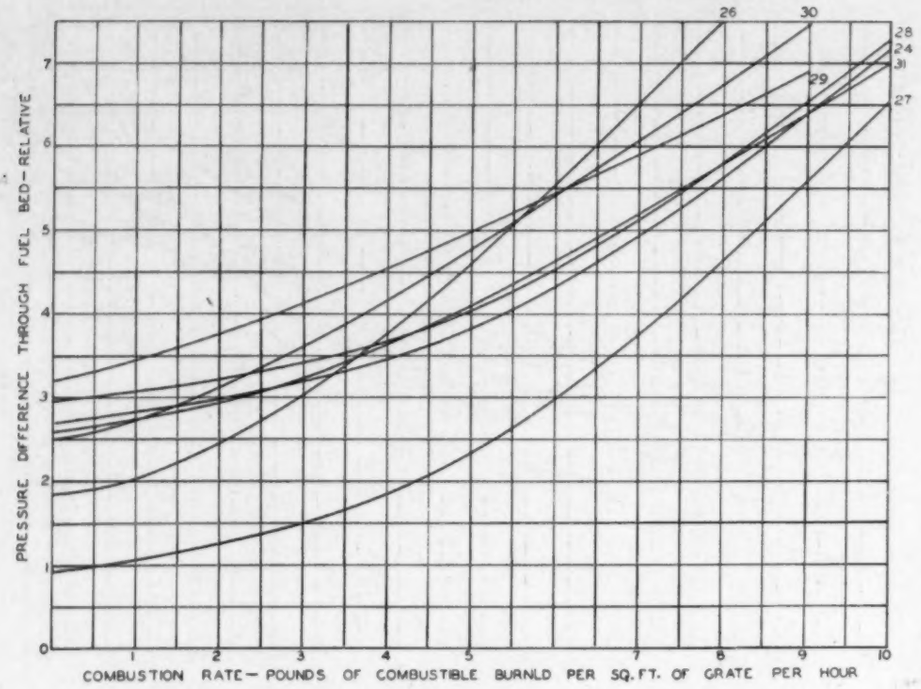

FIG. 5—CURVES SHOWING RELATION BETWEEN RELATIVE DRAFT PRESSURES AND RATE OF COMBUSTION FOR SAMPLES TAKEN FROM EACH VEIN MINED AT *W* COLLIERY

seen to be slower of ignition than anthracite No. 21, or even the anthracite of high draft requirement (No. 28*B*), while sample No. 28 falls to the lowest because of its high ash content. From these comparisons, it might be inferred that the ash particles act as an insulator against rapid ignition since on this chart the positions of the high ash samples No. 28 and the low ash No. 28*B* are reversed with respect to Fig. 1. Furthermore, Rhode Island graphite is found to have no horizontal ignition properties at all, thus barring it from use as a fuel. Coke, while having a slow initial horizontal ignition quickly overcomes the lead due to its greater proportion of voids.

In order to reflect the combustible nature of anthracite generally, samples were obtained from five different anthracite mines during the course of this investigation, the samples being so selected as to represent a general survey of the coal mined in each property. The graphs of the combustibility tests on the samples from three of the mines are shown in Figs. 3 to 8, inclusive.

The curves for the survey at colliery *X* (Figs. 7 and 8) show a very close uniformity between the samples from the various veins. This mine is favorably known because of the uniformity and free burning characteristics of its product.

Collieries *V* and *W* charted on Figs. 3 to 6 are similar to the great majority of anthracite mines and illustrate the general range of variability in this fuel. Colliery *W* is perhaps more troublesome in combustible characteristics because there are fewer veins and these few widely distributed.

The method of production now in vogue at these mines does not permit of blending except perhaps in an accidental way. When a train of mine cars is hauled from the mine to the top of the breaker there may be ten

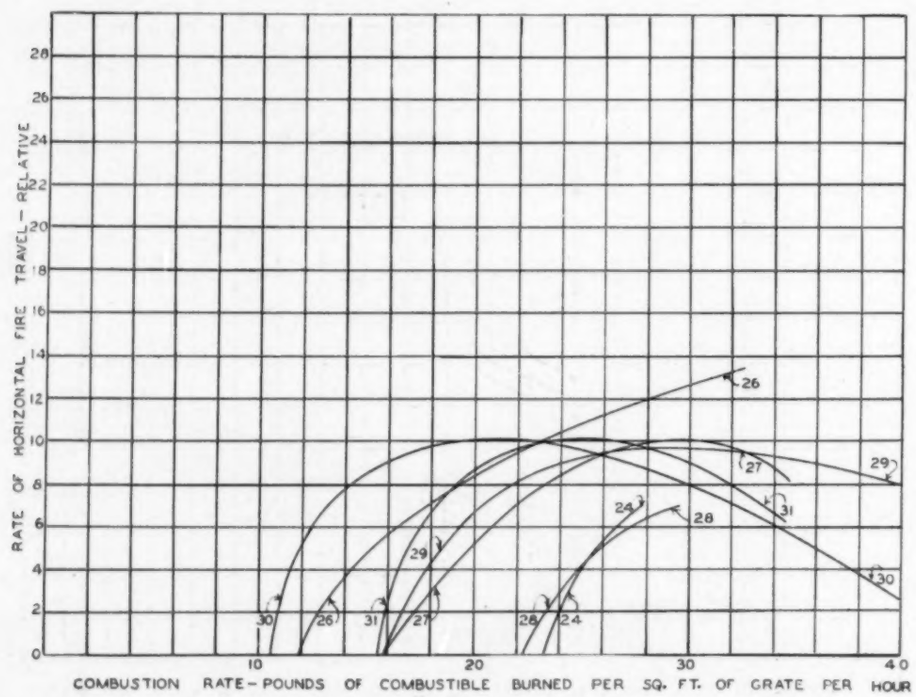

FIG. 6—CURVES SHOWING RELATION BETWEEN RELATIVE RATES OF HORIZONTAL FIRE TRAVEL AND RATES OF COMBUSTION FOR SAMPLES TAKEN FROM EACH VEIN MINED AT *W* COLLIERY

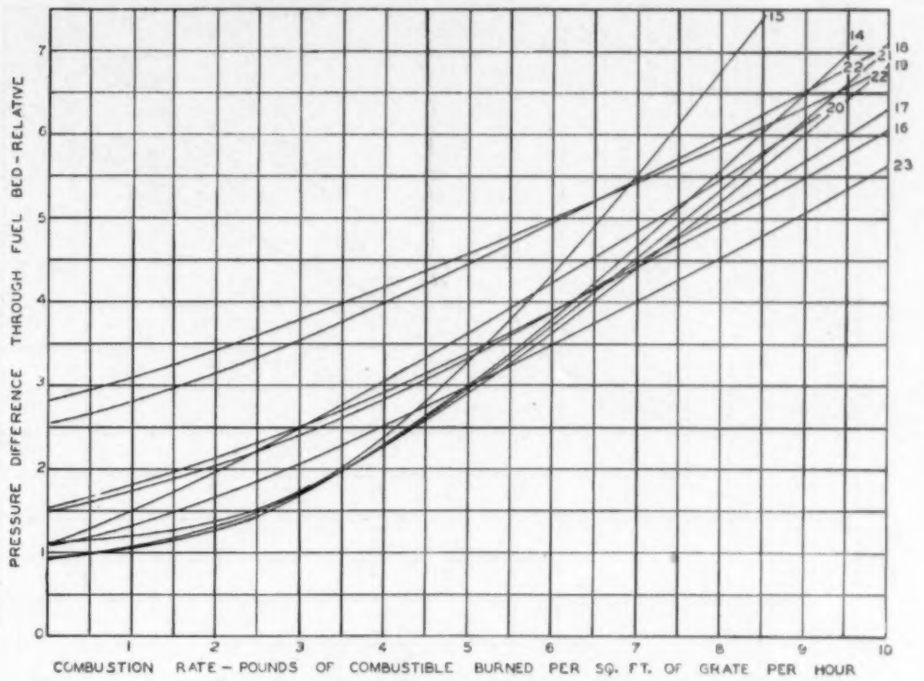
PRESSURE DIFFERENCE THROUGH FUEL BED - RELATIVE
COMBUSTION RATE - POUNDS OF COMBUSTIBLE BURNED PER SQ. FT. OF GRATE PER HOUR

FIG. 7—CURVES SHOWING RELATION BETWEEN RELATIVE DRAFT PRESSURES AND RATE OF COMBUSTION FOR SAMPLES TAKEN FROM EACH VEIN MINED AT *X* COLLIERY

cars all loaded at one locality in the mine. This coal passes through the breaker, is washed and sized and loaded into a railroad car entirely segregated from the coal from any other part of the mine. The coal loaded in a car may thus be either of the hardest or the freest burning coals or a hard burning coal may be deposited in the car followed by a free burning coal, resulting in a conglomerate mixture not uniformly blended.

That part of a consumer's heating equipment comprising the heater and chimney may not operate economically on a free-burning coal, or the plant may have an improperly-designed chimney, thus requiring a free-burning coal, or again the consumer may find he has received an unbalanced mixture, resulting first in a free-burning fire and then in a slow-burning fire. The outcome is annoyance and dissatisfaction, followed by a complaint to the dealer and transmitted by the dealer to the producer. Neither the dealer nor producer, having records of the combustible character of the fuel, is able to determine the cause of the trouble and continues in ignorance, repeating the same offense over and over. Clinkers, sizing, bone and slate and ash percentage are all blamed, whereas if the coal had been uniformly blended no complaint would have been made.

That proper blending will result in a very uniform coal is apparent from Figs. 9 and 10 which show curves of blended coals from the five different mines. These mines spread from one end of the anthracite field to the other, yet when the coals from each mine are properly blended the composite coal from any one mine is very similar in combustible character to the composite coal from any of the other four mines, the single exception

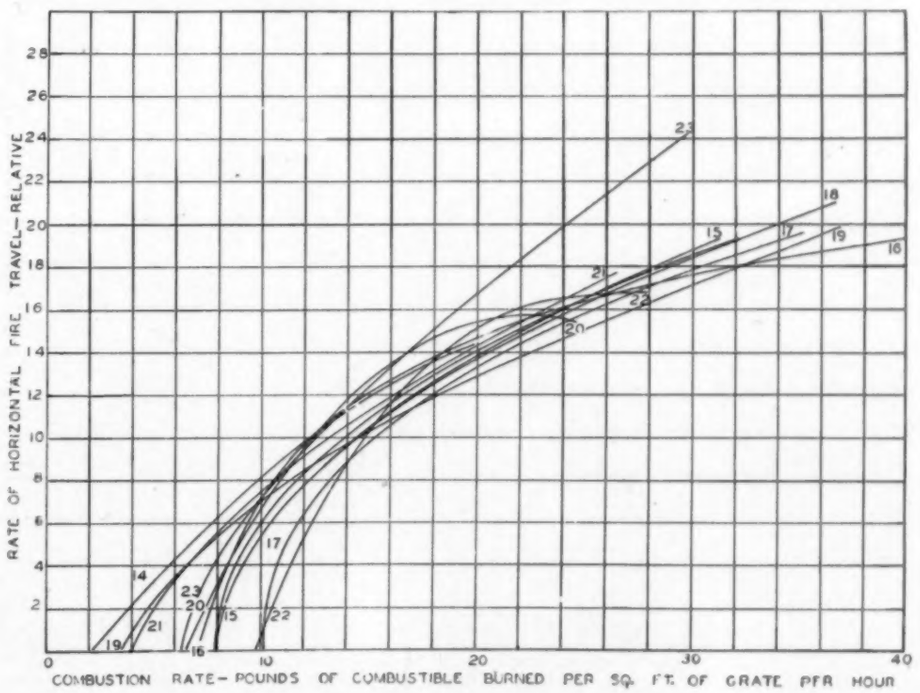

FIG. 8—CURVES SHOWING RELATION BETWEEN RELATIVE RATES OF HORIZONTAL FIRE TRAVEL AND RATES OF COMBUSTION FOR SAMPLES TAKEN FROM EACH VEIN MINED AT *X* COLLIERY

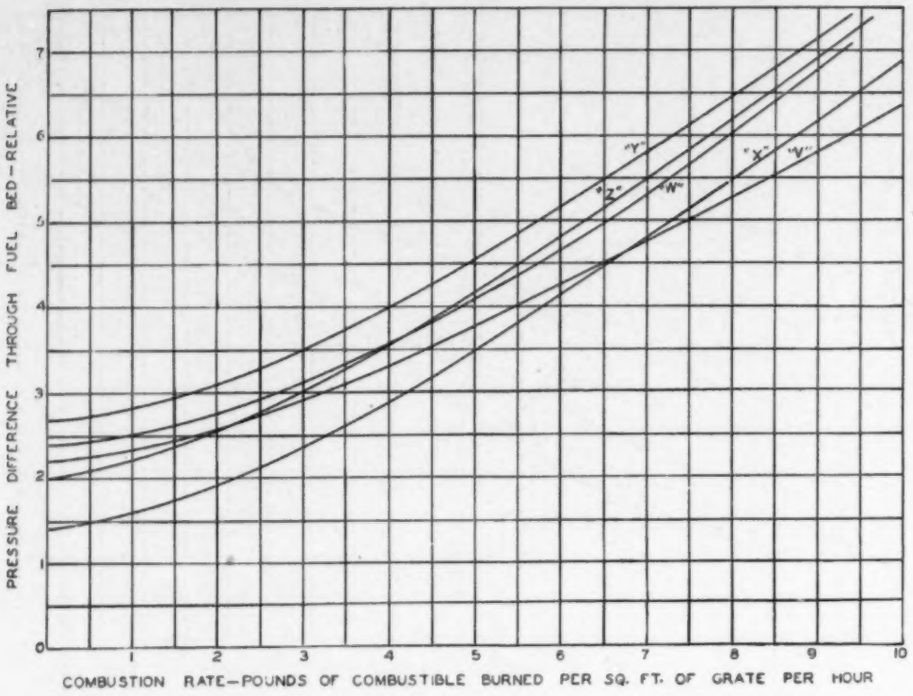

FIG. 9—RELATIVE DRAFT PRESSURES PLOTTED AGAINST RATE OF COMBUSTION FOR AVERAGE SAMPLES FOR EACH LOCALITY

being from mine *X*, where the coals are so uniform in character that the blend is still much freer burning than the blended coal from any of the others. Not only does the blend make a more uniform coal but it makes a coal that requires less draft, ignites more readily, and produces less clinker.

The reason the blended coal burns with less clinkers is explained by the fact that the curves of the coals having clinkering tendencies take a decided upturn at some point on the chart. When this takes place it indicates that the temperature of the fuel has reached a point that the ash sinters, thus closing off the admission of air to the carbon. In the blended coal the mixture of ash from the various veins tends to prevent the sintering of the various particles in the ash, perhaps due to the insulating effect of the non-clinkering ash which keeps the particles likely to fuse from reaching a critical fusion temperature.

Furthermore, it is clearly demonstrated that the ash structure is responsible for either the free or hard burning characteristic of a coal. This is explained by the fact that in the free burning coal the ash falls away from the carbon as it is formed thus permitting contact of the hot carbon with the oxygen of the air, whereas if the ash retains its form and continues to enclose the carbon the fuel is hard or slow burning. Observation has shown that it is very difficult to ignite the carbon through an ash exceeding one-half inch in thickness thus explaining why the coals of the fine prepared sizes can be burned to a more perfect ash than can the so-called senior or domestic sizes.

Another phase of the surveys of the five mines is shown in Figs. 11 and 12. It

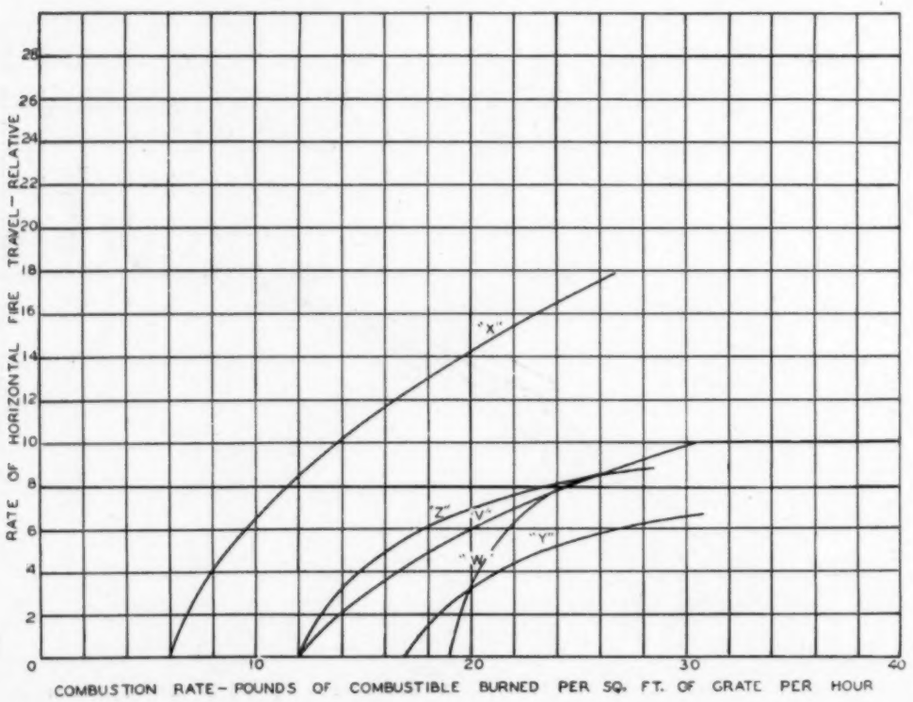

FIG. 10—CURVES SHOWING RELATIVE RATES OF HORIZONTAL FIRE TRAVEL PLOTTED AGAINST RATES OF COMBUSTION FOR AVERAGE SAMPLES FOR EACH LOCALITY

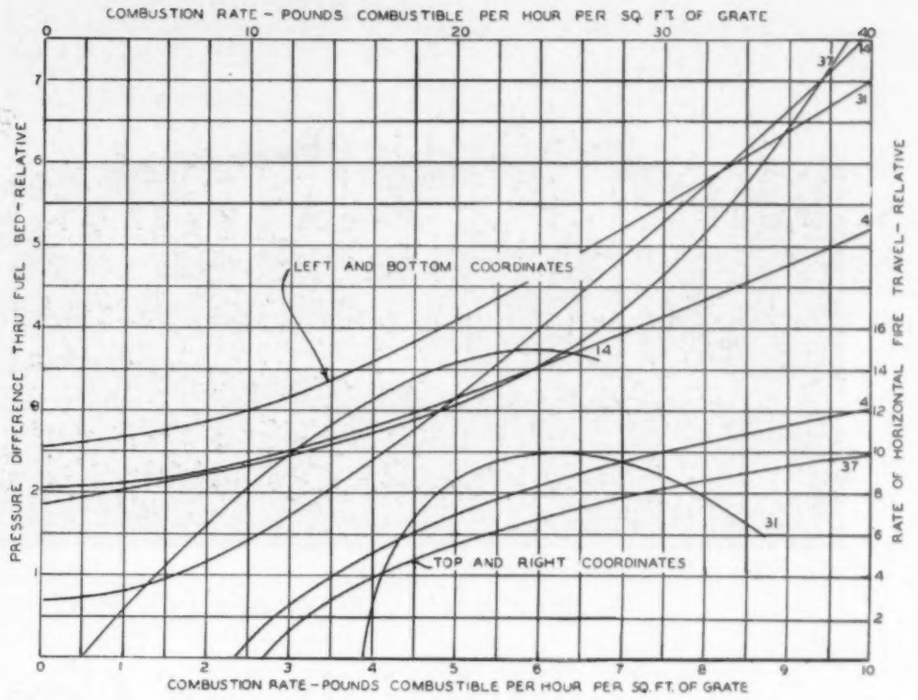
COMBUSTION RATE - POUNDS COMBUSTIBLE PER HOUR PER SQ. FT. OF GRATE
PRESSURE DIFFERENCE THRU FUEL BED - RELATIVE
RATE OF HORIZONTAL FIRE TRAVEL - RELATIVE
LEFT AND BOTTOM COORDINATES
TOP AND RIGHT COORDINATES
37
31
14
4

FIG. 11—COMBUSTIBILITY OF COAL VEIN *B* AT DIFFERENT LOCALITIES

is frequently claimed that coal from a particular vein is of unusual quality regardless of the section of the field in which the vein is located. That this claim is unfounded is illustrated by the curves of the same veins as they were found in the different mines. The wide spread of the graph lines shows the wide divergence in the combustible character of the fuel from the different areas. Coal from a certain vein in one mine may be very free burning while coal from the same vein in another mine may be extremely hard burning.

Conclusions

It is apparent that the usual tests of coal fail when applied to the determination of the combustible character of the coal. Neither appearance, density, chemical analysis nor location can be used as an indicator. The only positive means of determining the combustible character of a coal before it is put to use is to make a specific test for that particular purpose.

That the determination of the combustible character of the coal is highly essential to its proper utilization cannot be denied. Under hand-firing practice it was possible to blunder along with efficiencies of 50 or 60 per cent without serious complaint but with the increasing employment of automatic burning devices and the increasing use of coke, oil and gas as competitive fuels a coal that is not uniform in combustible character becomes a drag upon the entire practice of coal utilization, for with the absolute necessity of close adjustment and balance between air and coal feed in the automatic burning of coal, any coal that is not uniform in combustibility is immediately condemned.

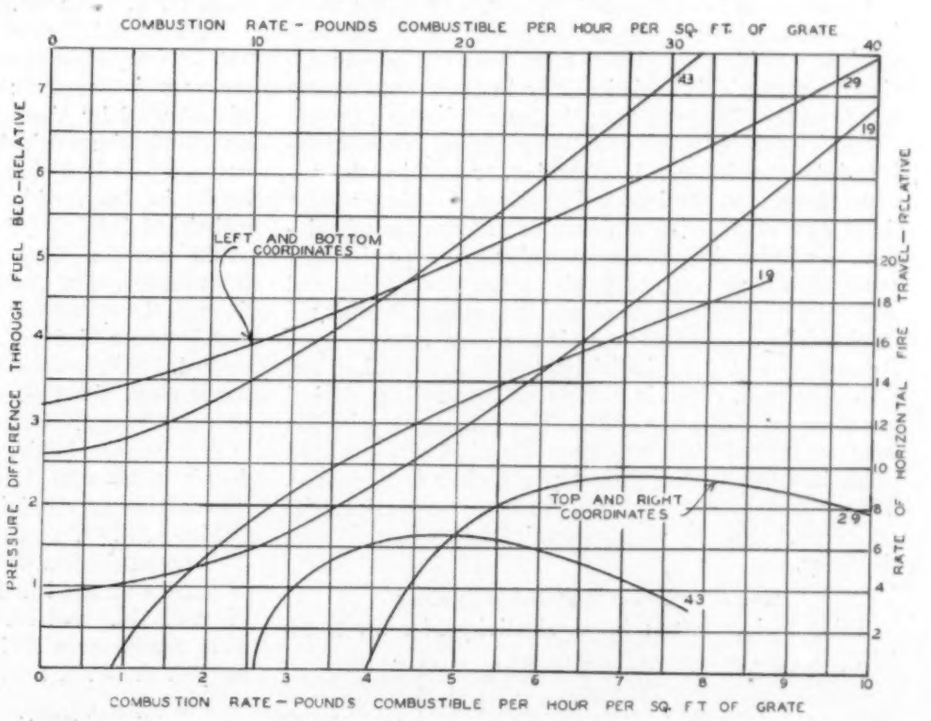
COMBUSTION RATE - POUNDS COMBUSTIBLE PER HOUR PER SQ. FT. OF GRATE
PRESSURE DIFFERENCE THROUGH FUEL BED - RELATIVE
RATE OF HORIZONTAL FIRE TRAVEL - RELATIVE
LEFT AND BOTTOM COORDINATES
TOP AND RIGHT COORDINATES
43
29
19

FIG. 12—COMBUSTIBILITY OF COAL VEIN *F* AT DIFFERENT LOCALITIES

PITTSBURGH EXPERIMENT STATION OF THE U. S. BUREAU OF MINES WHERE THE RESEARCH LABORATORY OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS IS LOCATED

Conductivity of Concrete

By F. C. Houghten[1] (*MEMBER*) and Carl Gutberlet[2] (*NON-MEMBER*)
Pittsburgh, Pa.

THE necessary design data for estimating heat loss from buildings have been greatly improved through research during past years. Probably the greatest need for further refinement is for additional facts concerning the variation in conductivity of concrete and stone masonry walls with mixture of ingredients, nature of the aggregate and ageing.

Investigators have reported heat transfer values varying over a wide range and indicating an average coefficient considerably higher than the most widely accepted value. This variation may in some measure result from the different test methods used, but to a greater extent it no doubt reflects the true variation in conductivity of the samples tested. Tamping, or working the wet concrete mortar, eliminates air voids and makes a more dense structure having a higher coefficient of conductivity. Variation in percentage and size of particles of cement, sand and aggregate also affect the number and size of air cells and the density of the mass. The thermal conductivity of concrete depends in a large measure upon the conductivity of the individual particles of sand, gravel or broken rock, which in turn depend upon the geological structure of these materials. The cement chemist has long recognized that important chemical changes take place in concrete for several years after it is poured. These changes affect its chemical composition and density and should logically be accompanied by changes in conductivity.

Results of Various Investigators

Table 1 gives values for conductivity of concrete reported by various investigations and shows values ranging from 2.3 to 17.4 Btu per square foot per hour per degree temperature difference per inch of thickness. The most widely accepted value is that reported by Willard and Lichty as 8.31.[a] This value has served as the basis for the recommendation of 8.0 and 8.3 for the conductivity of concrete used in tables in past issues of THE A. S. H. V. E. GUIDE.

Carmen and Nelson[b] in 1921 made a long series of tests on forty-eight concrete samples of six different mixtures ranging from 1:1.2:1.1 to 1:5.6:5.1. The method of test used by Carmen and Nelson was unique when compared with methods of other investigators. These tests were made on concrete cylinders, 7½ in. in diameter by 24 in. long. with an electrical heater placed in a cylindrical hole, 1½ in. in diameter, through the axis of the concrete. The conductivity values were calculated from the electrical energy input and the temperature drop through the cylindrical shell. Tests were made approximately one and four months after the cylinders were poured. Before each test, the sample was heated to a temperature above the boiling point of water to insure perfect dryness. The entire series of tests showed a variation in conductivity from 6.1 to 17.4 which bore no definite relation to either the mix or age of the samples. The average conductivity of all tests was 10.29.

The A. S. H. V. E. Research Laboratory, in making a survey of the heat flow characteristics and conductivity of walls with a Nicholls heat flow meter, reports values of 11.35, 11.8 and 12.5. The values of 11.35 and 11.8 were found for a 26-in. concrete wall in the basement story of the U. S. Bureau of Mines building. The value

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For presentation at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cleveland, Ohio, January, 1932.

a, b, etc. See Bibliography.

TABLE 1—CONDUCTIVITY OF CONCRETE ACCORDING TO VARIOUS INVESTIGATORS

AUTHORITY	MIX	AGE	THICKNESS INCHES	METHOD OF TEST	CONDUCTIVITY k			MEAN TEMPERATURE F
					AVERAGE REPORTED	HIGHEST VALUE	LOWEST VALUE	
A.S.H.V.E. GUIDE	1:2:4				8.30			
Willard and Lichty	1:2:4		3.19	Hot air box	8.31	8.74	7.89	
Norton[g]	1:2:4			Electric hot plate	8.42			2012.
Krueger and Eriksson[h]	1:5:0	105 Days	4.	Hot air box	9.4			36.77
"	"	"	8.	"	9.2			36.41
"	"	"	16.	"	9.5	9.6	9.4	37.13
Carmen and Nelson	1:1.2:1.1	28 Days	2.5 cylindrical shell	Electric input	9.72	12.48	6.97	
"	"	120 "	"	"	9.61	11.03	8.70	
"	1:1.9:1.7	28 "	"	"	11.55	13.93	9.00	
"	"	120 "	"	"	10.59	12.48	9.57	
"	1:2.4:2.3	28 "	"	"	10.92	13.64	7.55	
"	"	120 "	"	"	9.78	10.74	8.13	
"	1:3.1:3.0	28 "	"	"	9.58	13.06	8.13	
"	"	120 "	"	"	8.62	10.74	6.10	
"	1:4.3:4.0	28 "	"	"	11.03	12.77	7.55	
"	"	120 "	"	"	11.03	10.74	9.29	
"	1:5.6:5.1	28 "	"	"	9.87	11.03	6.97	
"	"	120 "	"	"	11.23	12.77	10.16	
"	"	150 "	"	"		17.41	11.03	
A.S.H.V.E. Lab	1:3:5	7 Years	26.2	Heat flow meter	11.35			44.
"	"	10 "	"	"	11.80			
"		4 "	9.0	"	12.50			
"	1:2:5	20 Days	7.94	"	16.26			80.
"	"	214 "	"	"	14.36			79.
"	"	3.4 Years	"	"	13.20			78.
"	1:2:4½		8.06	"	14.42			78.
"	"	142 Days	"	"	13.37			79.
"	"	3.8 Years	"	"	13.00			78.
"		1 Year	12.	"	7.08[a]			51.7
A.S.H.V.E. GUIDE			12.		6.22[a]			
Van Dusen					10.00			
Rowley	1:2:4	643 Days	5.95	Guarded hot box	11.90			39.95
"	1:2:3½	291 "	6.07	"	12.33			39.99
"	"	303 "	6.04	"	12.66			39.93

[a] 12 in. hollow concrete block—apparent conductivity including air space.
[g] [h] See Bibliography.

of 11.35 was reported [c] by Mr. Nicholls in 1923 for a section of the wall several feet above the ground level. The value of 11.8 was obtained by Mr. Zobel on a section of the wall, also several feet above the ground level, in another part of the building in 1926. The value of 12.5 was determined by the Laboratory for a 9-in. basement wall above the ground level in a four year old residence. Dr. Van Dusen of the United States Bureau of Standards reports a value of 10 for a single test on a wall for which there is no record of the mix or age other than that the aggregate was sand.

The A. S. H. V. E. Research Laboratory reports [d] a conductance of 0.59 Btu per square foot per hour per degree temperature difference from surface to surface for a 12-in. hollow concrete block wall. This gives an apparent conductivity of 7.08 Btu per square foot per hour per degree temperature difference per inch thickness for the entire 12-in. thickness including air space and concrete. This wall was in the basement of a residence which was one year old. However, the age of the blocks before laying is not known. THE A. S. H. V. E. GUIDE, 1931, gives the conductance of a similar wall as 6.22 Btu per square foot per hour per degree temperature difference from surface to surface. Prof. F. B. Rowley reports [e] conductivities ranging from 11.34 to 12.66 for three samples of concrete wall tested in a guarded hot box at the University of Minnesota.

The wide variation in conductivity of concrete reported by different investigators points to the desirability of making a survey of the variation in conductivity of concrete and masonry walls due to mix, character of aggregate used, and other factors which may be effective.

It is of interest to note that the value of 8.3 used in THE A. S. H. V. E. GUIDE, 1931, is about the lowest limit of those reported with the exception of a few values obtained for unusual test conditions. All of the data cited indicate an expectancy for a range in conductivity of concrete walls more than two years old of 8 to 14 and would further indicate that a value of 12 would be more commonly met with in practice than the value of 8.

In order to protect the interests of the designing engineer and contractor, as well as the owner, and to guard against failure, it would seem desirable to accept a value above the average rather than below, until such time as sufficient data may be made available to allow the intel-

[d] [e] See Bibliography.

ligent use of different values applying to different grades of concrete. A value of 12 is recommended.

Variation in Conductivity of Concrete with Ageing

Failure of well designed heating systems to satisfactorily handle the heating load of concrete buildings during the first and sometimes the second year after completion has lead many heating engineers and contractors to the belief that heat loss from concrete buildings is larger during this period than during subsequent years.

This lack of confidence on the part of the engineer and contractor, in the application of the design data for estimating the heat loss from concrete buildings during the first and second year, lead to an investigation by the Research Laboratory of the effect of ageing on the conductivity of concrete. A study of two concrete slabs, 8 in. thick by 2 ft square, was undertaken in 1927. Both slabs were well tamped and worked when poured. Slabs *X* and *Y* had cement, sand, gravel ratios of 1:2:5 and 1:2:4½ respectively, and in the spring of 1931 their densities were 137.5 and 139.9 pounds per cubic foot. Both slabs were housed in the A. S. H. V. E. Research Laboratory at Pittsburgh from the time of their pouring where they were subjected to the prevailing indoor atmospheric conditions.

The curves in Fig 1 show the variation in conductivity of the two slabs with time after pouring. Both show a very definite decrease in conductivity, the greatest rate of decrease taking place during the first two hundred days of the life of the slabs, but apparently continuing for more than three years, or until the day of their last test, when the samples were 1275 and 1405 days old respectively. The conductivity values are for mean temperatures between the warm and cold surfaces of the slab ranging from 77 to 80 F.

The different points show a variation from the mean curve as drawn, ranging to a maximum of 0.2 Btu or ± 1.5 per cent. This percentage is about double the percentage of error which would be anticipated due to the method of test. These errors must be accounted for, in part, as irregular variations in conductivity of the slab superimposed upon the progressive change shown by the curve, and may result from change in moisture content or other factors.

The two samples of concrete give curves showing different slopes or rate of change in conductivity with time, particularly during the last two years. There is no apparent reason for this difference. Sample *Y* was not tested as often as sample *X*, and little is known of its conductivity from 142 to 1289 days after pouring. A knowledge of changes taking place during this time might result in a curve of somewhat different shape.

The solid line curve best fitting the conductivity points for sample *X* shows an appreciable rate of change in conductivity for the present time, which if continued during subsequent years, would indicate a material effect on heat loss from concrete buildings throughout their life. However, it should be pointed out that the apparent irregular variation in conductivity from time to time, as indicated by the failure of the points to fit the curve more closely, may account for a considerable part of the apparent slope of the curve as drawn for the past two years. The dotted line curve for sample *X* shows another possible way of drawing the curve through the points with no greater maximum distance from any points to the curve than shown for the solid line curve. It would also predict a very different future for the conductivity of this sample. The difficulty in establishing the curve and its prediction for the future from the few points available indicates the desirability of occasional tests on these samples during the next several years, or until the change in conductivity with time can be shown to have ceased.

During the first year after pouring, the respective

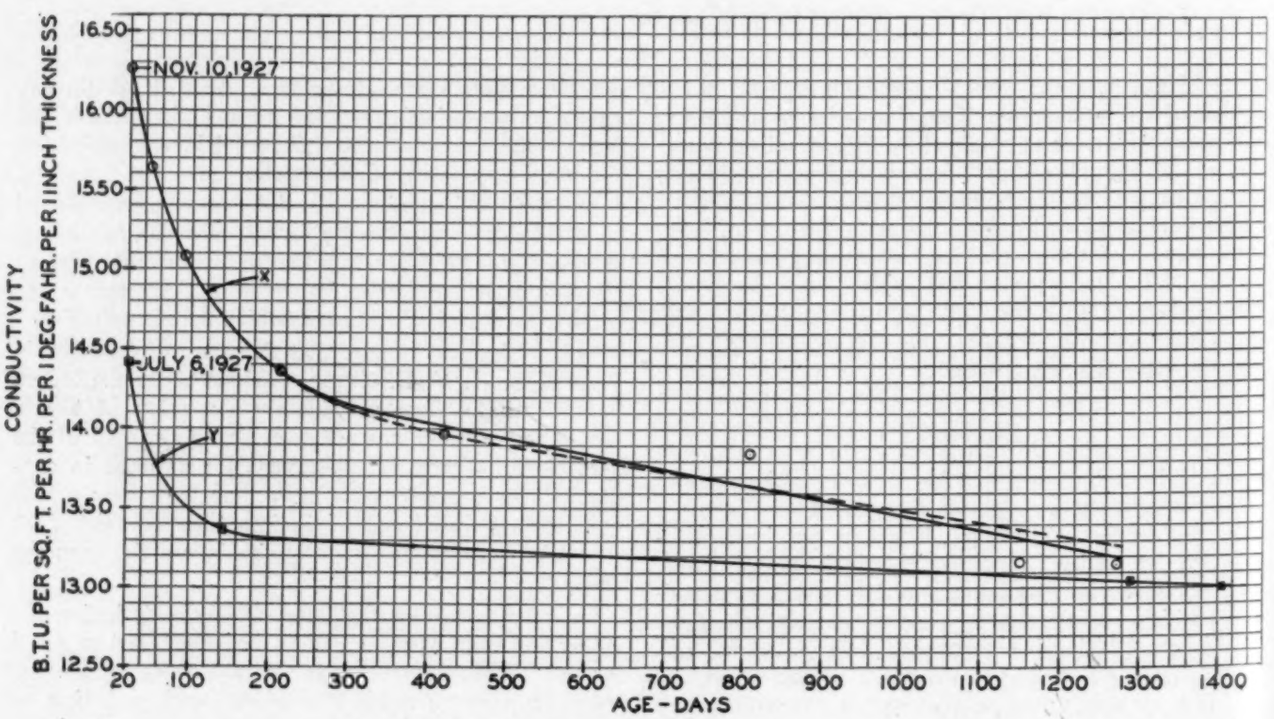

FIG. 1—VARIATION IN CONDUCTIVITY OF CONCRETE WITH AGE

samples *X* and *Y* showed a decrease in conductivity of approximately 13 per cent and 8 per cent. From the end of the first year to the time of the last test, the decrease amounted to approximately 6 per cent and 2 per cent, or an average yearly rate of decrease after the first year of 2.5 and 0.7 per cent. The total percentage decrease in conductivity since pouring is 18.8 and 9.8 per cent for the respective samples.

The higher conductivity of concrete indicated for the first and second years after pouring would result in a very noticeable increase in heating load during the first season, and some increase the second season, over subsequent years for a building whose walls consist largely of concrete without insulation or finishing on the inside. It should be noted, however, that for walls consisting of concrete and some other material, having a considerable higher resistance to the flow of heat, the effect of variation in conductivity of concrete with age would have a smaller percentage effect on the heating requirement for the first year.

It should be noted that the conductivity shown by these two samples is higher than any other value listed in Table 1 with the exception of certain tests by Carmen and Nelson. The only apparent reason for this is that these samples were probably worked and tamped more during pouring than is the usual practice.

Method of Tests to Determine Variation of Conductivity of Concrete with Age

Fig. 2 is a photograph and Fig. 3 a drawing of the test apparatus. The samples were tested in the double guarded hot plate designed and built at the A. S. H. V. E. Laboratory by P. Nicholls.[†] *A* is a 2-ft square, main guarded hot plate. *B* is an auxiliary guarded hot plate, separated from *A* by a high-resistance, heat-flow meter *C* approximately ½ in. in thickness. In operating the double guarded hot plate, the electric current through the center and also that through the guard ring of heater, *A*, are adjusted to allow no temperature difference between the center and guard ring. Heater *B* is likewise adjusted to give no temperature difference between its center and guard and at the same time no heat flow through *C*. With this condition prevailing, it is obvious that all heat generated in the center heater of *A*

[†] See Bibliography.

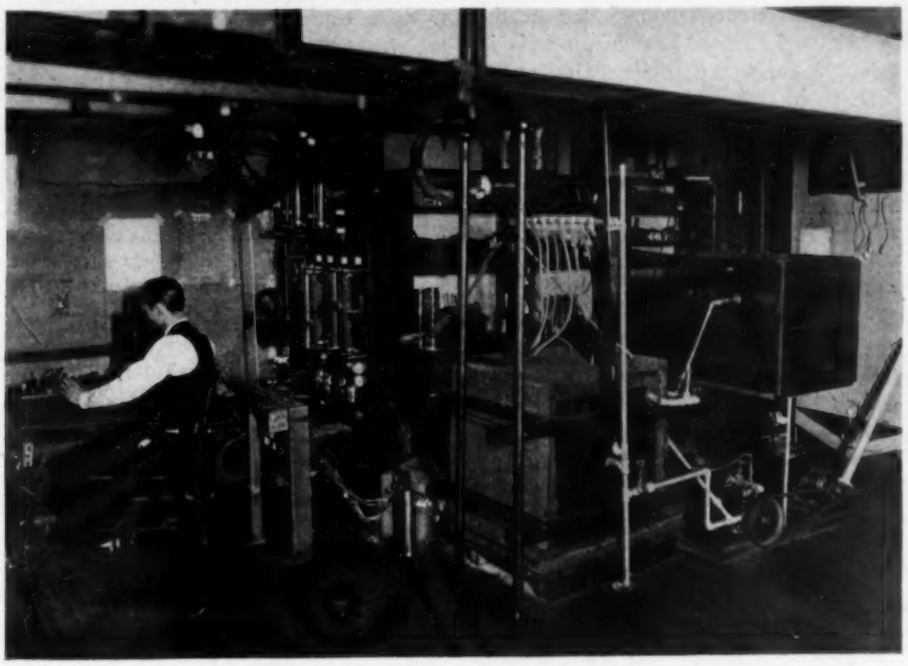

FIG. 2—APPARATUS FOR DETERMINING THERMAL CONDUCTIVITY

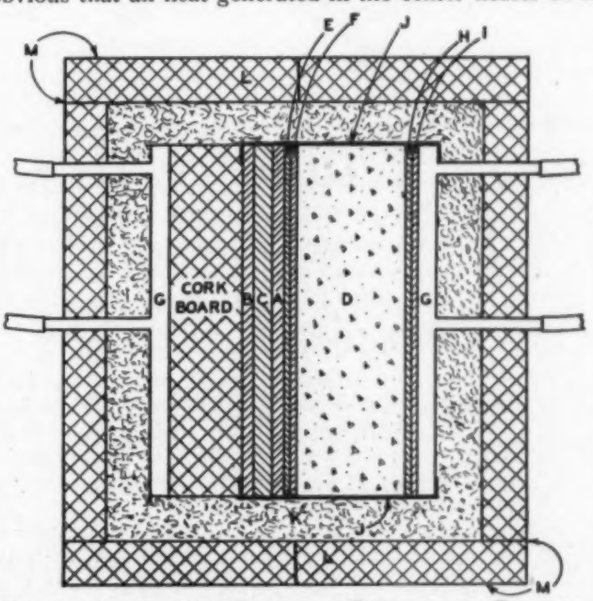

FIG. 3—GUARDED HOT PLATE FOR DETERMINING THERMAL CONDUCTIVITY

A—Main hot plate.
B—Auxiliary hot plate.
C—High resistance heat meter.
D—Concrete test sample.
E & *F*—Heat meters, warm side.
G—Water cooled heat absorber.
H & *I*—Heat meters, cool side.
J—Wax paper.
K—2 in. hair felt.
L—2 in. corkboard.
M—Waxed surface and joints.

must flow in the direction of D. D is the concrete slab under test with two low-resistance Nicholls heat flow meters, E and F, between it and the main heater. G is a copper, water-cooled, heat absorber and H and I are two Nicholls heat flow meters similar to E and F.

The water circulating through the water-cooled plate, G, is thermostatically controlled to ± 0.05 deg of that desired. In order to guard against heat loss by air infiltration through the system, wax paper is sealed around the perimeter of the entire assembly which is then insulated with two inches of hair felt and two inches of cork board. With this equipment, the rate at which heat was supplied to the warm side of the sample could be measured by the electrical input of heater A or by heat meters E and F. The heat dissipated from the cold side of the concrete was measured by meters H and I.

In order to minimize error due to stray heat flow, the room temperature was maintained at the mean of the two sides of the concrete slab. The temperature drop through the concrete slab was determined by five thermocouples placed on each face. Even with this precaution, the heat recovery at the cold side of the concrete slab as measured by H and I was always less than that supplied at the hot side and measured by meters E and F. This is a fact worthy of consideration in the designing and application of the usual hot plate method to the determination of conductivity of thick slabs of materials having a high conductivity. The usual method takes no cognizance of this loss, the customary practice being to make the hot plate large enough so that this loss may be assumed to be negligible.

In calculating the conductivity, the average heat flow indicated by the meters on the warm and cold sides and the temperature drop through the concrete slab were used in the usual formula:

$$k = \frac{qL}{A(t_1 - t_2)}$$

where

k = thermal conductivity (heat transferred in Btu per square foot, per hour, per degree difference F per inch of thickness).

q = heat transferred per unit time (Btu per hour).

L = thickness of path of heat flow (inches).

A = Area (square feet).

t_1 = temperature, warm side (F).

t_2 = temperature, cold side (F).

Summary and Conclusions

This report points out some of the reasons for wide variation in conductivity of concrete and indicates that the value commonly used is probably too low an average to properly protect the practice of the consulting engineer and contractor. An average value of 12 is recommended until more is known concerning the effects concrete characteristics have on the conductivity. The decrease in conductivity of two slabs of concrete with age is shown by curves to be of considerable magnitude and progresses at a decreasing rate for at least 30 months after pouring.

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[b] University of Illinois Engineering Experiment Station Bulletin No. 122.

[c] Practical Applications of the Heat Flow Meter, by Percy Nicholls (A. S. H. V. E. TRANSACTIONS Vol. 30, 1924).

[d] Additional Coefficients of Heat Transfer as Measured Under Natural Weather Conditions, by F. C. Houghten, Carl Gutberlet and C. G. F. Zobel (A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929).

[e] To be published later.

[f] Measuring Heat Transmission in Building Structures and A Heat Transmission Meter, by Percy Nicholls (A. S. H. V. E. TRANSACTIONS, Vol. 30, 1924).

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A TRIBUTE TO A. S. H. V. E. RESEARCH

IT SHOULD be gratifying to members of the Society to know the esteem in which the Research efforts of the Society are held by heating and ventilating engineers of Great Britain, as expressed editorially in the June 1931 issue of *Domestic Engineering, Heating and Ventilation,* London, Eng., the official organ of the *Institution of Heating and Ventilating Engineers.*

RESEARCH IS NECESSARY

The Prince of Wales, on his return from South America, delivered some home truths to British business men and manufacturers when he explained some of the reasons for our loss of trade. He formed the impression that some of our manufacturers adhered too closely to pre-war types of manufacture.

America is making great advances due to their realization of the value of research. The AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has been developing its research work for some years, whereas our Institution in 1922 could not find sufficient support to enable it to carry on its research work, which had been doing useful work so far as finance would allow. With the closing down of the Institution's research work at University College, the industry was deprived of a valuable source of technical knowledge, but fortunately the Department of Scientific and Industrial Research undertook to carry out research work on heating and ventilation with the assistance of the funds provided for Building Research. The Institution has now another opportunity of showing its interest in the advancement of the technical side of the industry by its cooperation with the Department of Scientific and Industrial Research Standing Advisory Committee on Heating and Ventilation formed in 1928.

The American Society has just developed a new and comprehensive policy obtained from the experience of ten years of cooperative research and claims the distinction of being the only professional engineering organization to maintain and operate its own Research Laboratory and is now contributing 40 per cent of its annual subscriptions to scientific and practical value. That keenness of our sister Society shows the way of progress. Is it not time that we emulated their endeavors? The opportunity is available and it is hoped that the appeal which the Institution now makes will be read and acted upon by all those interested in heating, ventilation and allied work.

Experiments to Determine Heating Effect Factor

By R. N. Trane[1] and C. J. Scanlan[2], LaCrosse, Wis.
MEMBERS

CONVECTION-TYPE heaters are rated by most manufacturers in terms of square feet of equivalent direct "radiation" (240 Btu per hour) and the performance of these heaters usually is compared with that of exposed or direct cast-iron "radiation." However, shortly after the introduction of this type of heating equipment, it became apparent that equivalent temperatures could be maintained within the living zone of a room with less condensate when properly designed convection heaters were used than when exposed cast-iron "radiation" was used. This reduction in heat input to do the same heating work has been termed *heating effect.*

This difference was accounted for by the fact that floor temperatures were higher and ceiling temperatures were lower with convection heaters than with direct heaters. Although this claim was in a measure substantiated, it was impossible entirely to account for the large difference in condensation requirements between a good convection heater and an exposed cast-iron "radiator" by means of the small temperature differences existing within the room itself. Thus, although a properly designed convection heater usually will deliver a greater quantity of the available heat to the living zone of a room, temperature differences alone apparently do not entirely explain this *heating effect.*

Convection heaters, as well as exposed cast-iron "radiators" are now rated in terms of the square foot of equivalent direct "radiation" (240 Btu per hour) based on condensation tests run in still air at a temperature of 70 F. A convection heater must of necessity be entirely different from an exposed cast-iron "radiator," both structurally and physically.

Although an exposed cast-iron "radiator" may actually emit 240 Btu per hour per square foot of surface under standard conditions of 215 F (steam) and 70 F (air), the most efficient convection heaters have a very small amount of prime surface and a great deal of extended surface, and the heat emission therefore has no such relation to the actual heating surface.

Efforts have been made to combine *heating effect* with the rating of a convection heater. A code for testing and rating convection heaters for both steam and hot water is now under preparation by a committee of the Society. This code, when completed, should provide an accurate and correct basis for determining the heat output of a convection heater at the outlet grille. The important question, however, is—*What becomes of the heat delivered from this outlet grille in an actual installation?*

Purpose of Investigation

One purpose of the investigation reported in this paper was to determine why convection heaters require less heat input to maintain the same air temperature within the room than exposed cast-iron "radiators" when the room is exposed to identical cooling conditions in both cases. Another was to ascertain whether *heating effect* should be combined with rating or whether it is a function of some physical factor. In other words, the primary object was properly to evaluate the Btu input to a convection heater as compared with that of the various exposed cast-iron "radiators."

Laboratory Equipment

In order to accomplish this purpose, it was necessary to build a cold room of a type adaptable to the desired testing methods. A study was made of the cold room used at the University of Illinois, and also at the Frost Research Laboratories. In considering a cold room similar to the type used at the University of Illinois, the infiltration through windows and doors seemed to complicate the problem, in addition to the difficulty of maintaining similar conditions for a number of tests. It was therefore decided to build a room which eliminated infiltration and one having walls that could be cooled to any desired temperature and maintained at that temperature over an indefinite period. In other words, it was necessary that this room be so constructed and equipped that constant temperatures could be maintained for a period of eight hours or more if desired. Thus, infiltration and variable temperatures were eliminated, although it was realized that in practice infiltration cannot be disregarded.

The cold room used contained a heating space 8 ft wide by 12 ft long by 8 ft 3 in. high. Three walls were constructed of sheet metal and were painted on the interior the same as interior walls would be painted. Four thermocouples were soldered on the surface of each of these walls so as to obtain surface temperatures with rapidity and accuracy. A thermocouple was located 3 in. above the floor, at the breathing line, and 3 in. below the ceiling, approximately in the center of the room. A thermocouple was also placed immediately over the location of the heater at a distance of 3 in. from the ceiling and 3 in. from the wall. Two thermocouples were installed 2 in. above the floor and opposite the inlet to the heater, and two thermocouples were also provided for measuring the outlet temperature from the heater. The system was constructed in such a way that all thermocouples were balanced. All lead wires were made the same length and from the same spool of wire. The readings were taken by means of a milivolt meter.

A five-ton CO_2 compressor was located in the room beneath the laboratory, and was used for cooling water in an 800-gal tank. This water was in turn pumped through the nozzles in the spray chambers behind each of the sheet metal walls.

Figs. 1 to 7 illustrate the equipment used.

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Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1931.

FIG. 1—FRONT OF COLD ROOM

Procedure

Thirty-eight tests of 8-hours duration were conducted to determine the operating peculiarities and characteristics of the equipment before attempting to establish the proper procedure for operating the cold room. It was desired to ascertain whether or not stabilized conditions could be maintained over a period of hours. It was found that a point of equilibrium could be obtained when the heat input to the room exactly balanced the heat output.

The equilibrium was obtained in all cases by adjusting the size of the exposed cast-iron "radiator" or convection heater to the heating requirements. A smaller exposed cast-iron "radiator" was required with a 62-F wall temperature than was required with a 54-F wall temperature. Likewise, shorter convection heaters were required with a 62-F wall temperature than with a 54-F wall temperature.

After this preliminary work had been completed, it was decided that in order to obtain a comparison of the relative heating effects of the various exposed cast-iron "radiator" and convection heater tests it would be necessary to establish a definite temperature at some point within the room. A breathing line temperature of 72 F was therefore selected. It was also decided to maintain a definite cooling effect in each test, and one of the side walls and the end wall were selected for this purpose. These two cold-wall surfaces comprised an area of 166 sq ft, and several tests were conducted with surfaces maintained at temperatures of 54 F, 58 F, 62 F and 66 F.

These wall surface temperatures were selected because they would give results comparable to outside temperatures ranging from —20 F to +30 F according to the results obtained at the University of Illinois Research Residence, as reported in the paper *Wall Surface Temperatures* by A. C. Willard and A. P. Kratz.[3]

Test Results

The test results are given in Table 1. Attention is called to the fact that constant breathing line temperatures were maintained throughout the investigation, and that the wall temperatures were varied. All of the tests reported in Table 1 are for a cold wall having a surface area of 166 sq ft.

Discussion of Results

Because of the fact that equilibrium was maintained during all of the tests, it follows that the heat input to the cold room equals the heat loss. Therefore, inasmuch as the cold surface remained the same in all cases, not only in area but in temperature, and the inner air temperatures remained the same, the only factor of the room that could change was the coefficient of transmission of the wall (U). Fig. 8 indicates that with a wall surface temperature of 58 F, an exposed cast-iron "radiator" required an input of 3,440 Btu per hour to maintain a breathing line temperature of 72 F. Therefore:

U_{CI} = Overall coefficient of heat transmission of wall for cast-iron "radiator"

$$= \frac{3440}{166 \times (71 - 58)} = 1.59$$ Btu per hour per square foot per degree difference between the wall temperature and the average room temperature.[4]

[3] See JOURNAL of the Society (Heating, Piping and Air Conditioning), April, 1930.

[4] The overall coefficient of heat transmission for the test wall meets the definition of surface coefficient. For test conditions the wall surface temperature is also the temperature to which heat flows. In practice the walls have greater resistance than the sheet metal wall tested, but this does not change the fact that it is the overall coefficient of heat trans-

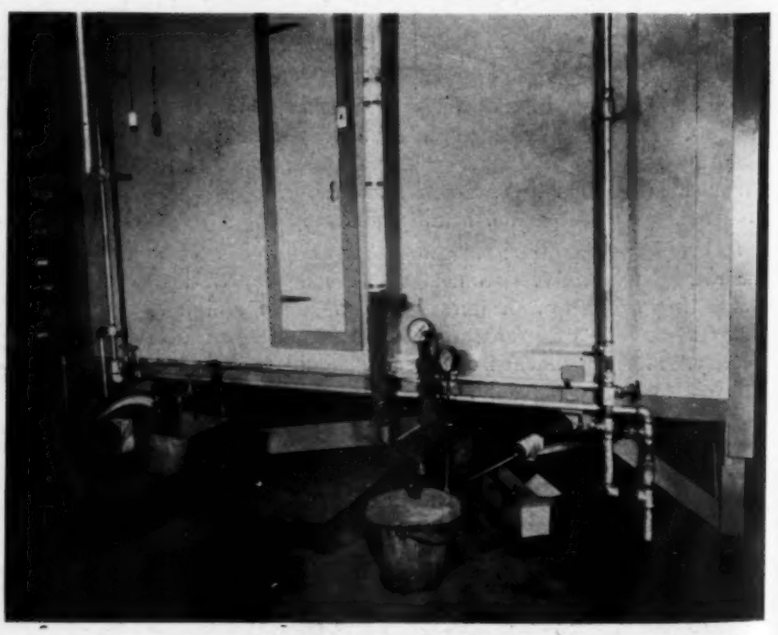


FIG. 2—REAR OF COLD ROOM

For a similar wall surface temperature, the convection heater required an input of 2,400 Btu per hour. Therefore:

U_c = Overall coefficient of heat transmission for convection heater

$$= \frac{2400}{166 \times (71 - 58)} = 1.11$$ Btu per hour per square foot per degree difference between the wall temperature and the average room temperature.

By means of these coefficients, the heat input required can accurately be predetermined. The *heating effect* is therefore equal to

$$\frac{U_c}{U_{cl}} = \frac{1.11}{1.59} = 0.7$$

A comparison of floor-line and breathing-line differences, or floor-line and ceiling differences, for the exposed cast-iron "radiator" and the convection heater tested cannot possibly account for this difference in heat input required to maintain the cold room at 72 F.

The apparent reason for this reduction in heat input

mission (U) that is affected by the source of heat supply. The test wall at 58 F with a cast-iron "radiator" passes 20.7 Btu per square foot per hour, which checks a standard wall having a coefficient of transmission (U) of 0.25 Btu per square foot per hour per degree which for −10 F to 70 F gives 20 Btu per square foot per hour heat loss. The convection heater with a 58 F wall passes 14.49 Btu per square foot of wall area.

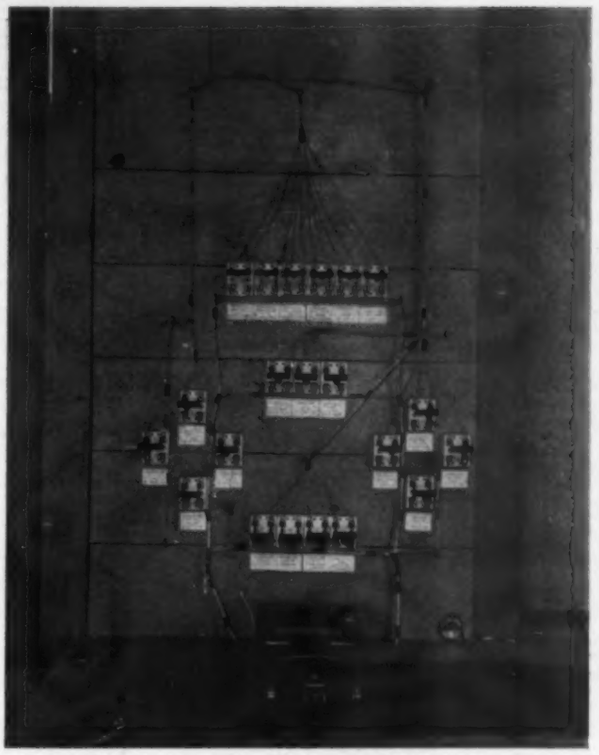


FIG. 3—THREE THERMOCOUPLE SWITCHBOARD

TABLE 1—RESULTS OF TESTS

Test Number	Equipment Tested	Floor Temp.	Breathing Line Temp.	Ceiling Temp.	Ave. Living Zone Temp.	Ave. Room Temp.	Ave. Wall Temp.	Air Temp. Entering Heater	Air Temp. Leaving Heater	Difference Floor to Breathing Line	Difference Floor to Ceiling	Pounds Condensate	Btu
1	5 Tube—38 in.—N. and W. Walls	63.5	72	79.6	68.0	71.8	54	73.8	163.1	8.5	16.1	4.34	4200
1	" N. and W. "	65.0	72	79.0	68.0	72.0	58	70.0	169.0	7.0	14.0	3.69	3580
1	" S. and W. "	67.2	72	76.2	69.4	72.0	62	72.8	169.4	4.8	9.0	2.50	2425
2	Convection Heater *A*	65.0	72	75.0	69.0	71.0	58	71.0	148.0	7.0	10.0	2.47	2400
2	" " "	67.0	72	74.8	69.6	71.3	62	73.0	128.0	5.0	7.8	1.94	1880
2	" " "	69.0	72	74.3	70.2	71.6	66	75.7	142.6	3.0	5.3	1.10	1067
3	Convection Heater *B*	63.6	72	76.5	67.8	70.7	54	69.5	120.0	8.4	12.9	3.39	3285
3	" " "	66.0	72	76.0	69.0	71.2	58	75.0	106.8	6.0	10.9	2.56	2480
3	" " "	66.0	72	75.2	69.0	71.1	62	75.6	101.2	6.0	9.2	1.61	1560
3	" " "	67.1	72	75.2	70.0	71.5	66	79.0	95.5	4.9	8.1	1.07	1038
4	Convection Heater *C*	63.2	72	75.0	67.7	70.1	54	68.2	130.6	8.8	11.8	3.30	3200
4	" " "	65.8	72	76.5	68.9	71.4	58	71.0	124.8	6.2	10.7	2.51	2420
4	" " "	68.3	72	74.0	70.1	71.3	62	75.8	128.7	3.7	5.7	1.56	1515
4	" " "	69.0	72	74.8	70.5	71.9	66	76.8	104.0	3.0	5.8	1.18	1140
5	5 Tube—26 in. Heater	65.5	72	77.2	68.6	71.5	54	72.1	145.0	6.5	9.7	4.10	3980
5	" "	65.8	72	77.1	69.1	71.8	58	73.0	180.0	6.2	11.3	3.58	3470
5	" "	67.0	72	77.0	69.5	72.0	62	75.0	173.0	5.0	10.0	2.95	2860
6	Convection Heater *D*	64.3	72	75.9	68.1	70.7	54	68.8	136.6	7.7	11.6	3.41	3300
6	" " "	65.1	72	75.3	68.3	70.6	58	69.2	130.5	6.9	10.2	3.01	2920
6	" " "	66.7	72	75.0	69.4	71.3	62	72.2	118.0	5.3	8.3	1.91	1850
6	" " "	68.7	72	74.1	70.3	71.6	66	75.0	129.0	3.3	5.4	1.03	1000
7	5*B*—Wall Section—4 Sections	65.8	72	75.0	68.8	70.9	54	77.4	146.5	6.2	9.2	4.38	4250
7	" " "	66.8	72	74.7	69.2	71.1	58	79.1	146.4	5.2	7.9	3.40	3300
7	" " "	67.8	72	73.7	69.8	71.1	62	79.3	159.5	4.2	5.9	2.35	2280
7	" " "	68.8	72	74.2	70.4	71.7	66	75.6	151.0	3.2	5.4	1.75	1700

FIG. 4—INTERIOR OF COLD ROOM

required by the convection heater compared with that of the exposed cast-iron "radiator" is that in one case the heat emitted from the exposed cast-iron "radiator" is made up of convected heat and radiant energy, while in the other the heat emission is almost purely convected heat. Inasmuch as convected heat only is capable of raising the air temperature, it follows that in the case of the particular room tested the percentage of radiant energy emitted by the exposed cast-iron "radiator" was 30 per cent as compared to the convection heater tested. No two convection heaters of different design will have the same *heating effect* coefficient.

Fig. 8 shows average results obtained for all of the tests. No material difference could be determined between *heating effect* coefficients for high and low exposed "radiators." Likewise, no appreciable difference could be detected between a convection heater with a high stack and one with a low stack. The only restriction in all cases is that of controlling the heat output of the convection heater or exposed cast-iron "radiator" to exactly equal the heat loss from the room.

It may be difficult to appreciate that these percentages are as large as herein reported. However, in Fig. 11 of a paper entitled Determination of Radiant Energy Given Off by a Direct Radiator, by John R. Allen and Frank B. Rowley[5], the following percentages are given:

No. 1—Plain Cast-Iron—Rusty.....38.3 per cent
No. 2—Aluminum Paint .....25.0 per
No. 3—Black Paint .....35.4 per
No. 4—Plain Cast-Iron—No Rust.....31.2 per cent

Bulletin No. 120 (March, 1921) entitled Investigation of Warm Air Furnaces and Heating Systems, published by the University of Illinois Experiment Station, Fig. 43-A, on page 114, shows that coefficient of emissivity for bright tin, not insulated, to be 1.28 Btu per square foot per hour per degree Fahrenheit; whereas, for black iron, No. 28 U.S.S. gage, very rusty, it is 2.37. Inasmuch as the surfaces of both materials were subjected to the same steam and air temperatures, it follows directly that for the black iron specimen 46 per cent of the heat was given off in the form of radiant energy.

Figs. 23 and 25 of University of Illinois Experiment Station Bulletin No. 223 (March, 1931), indicate the results of tests with practically the same breathing-level temperature. If the test on enclosure No. 3 (test E-la) is compared with the test on unenclosed "radiator" (test R-5b), the *heating effect* will be found to be 0.855. Also, in Fig. 25 of this bulletin, the *heating effect* factor for test E-3a as compared with test R-5b is 0.862. The heaters referred to in these tests are primarily radiators and are as a result poor convectors. Stream line air flow and proper heating surface contacts have been given little consideration. If, as reported in these tests, the addition of a cabinet or enclosure produces a factor of approximately 0.85, it follows that a good convector, scientifically designed, will give without question the *heating effect* factor of 0.7 herein reported. However, this factor will not be the same for convectors of different design.

In order to illustrate the manner in which this *heating effect* coefficient is used in the practical application to heat loss calculations, the following problem, involving a warm air installation with no radiant energy[6], is submitted:

[5]See A. S. H. V. E. TRANSACTIONS, Vol. 26, 1920.

[6]The data were taken from Bulletin No. 189, University of Illinois Experiment Station, Daily tests No. 105 and 111, page 26, corrected to a mean outdoor temperature of zero.

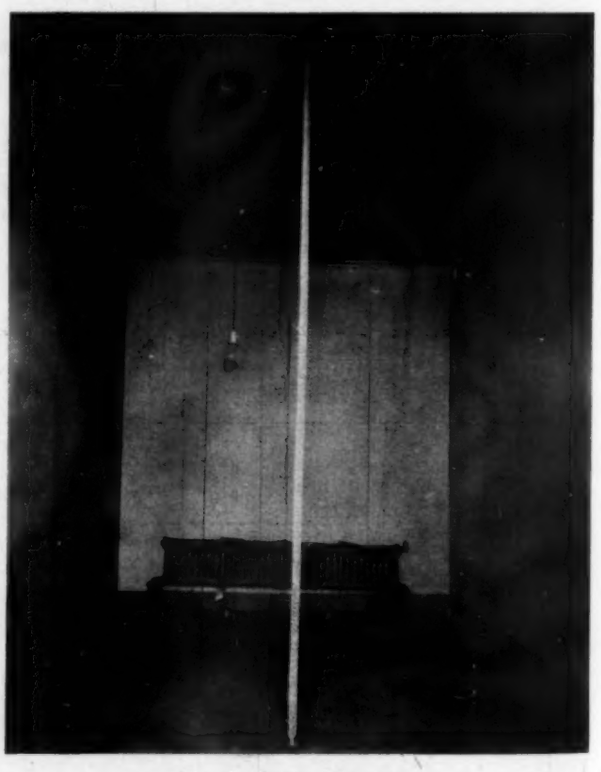


FIG. 5—SPRAY CHAMBER

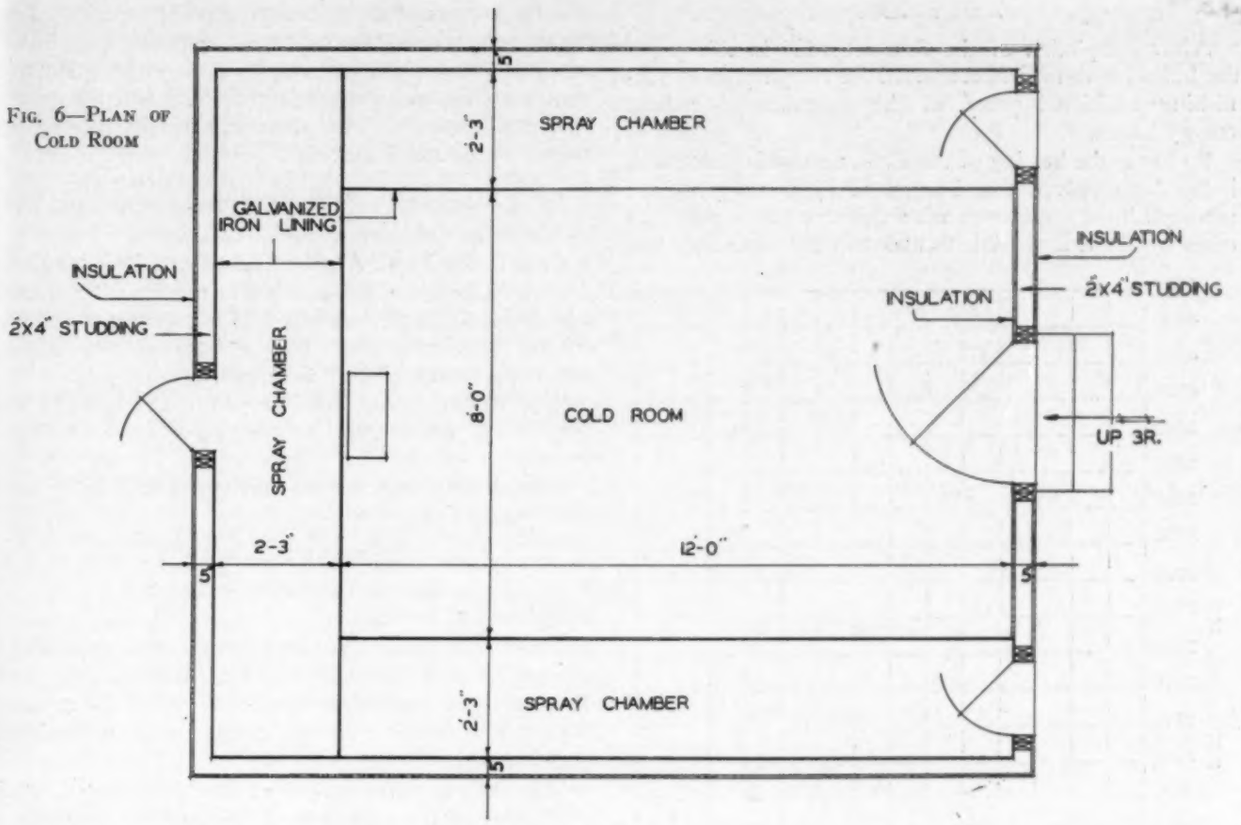


FIG. 6—PLAN OF COLD ROOM

Calculated heat loss (0-70 F)	140,000 Btu
Infiltration loss	29,000 Btu
Wall and glass loss	111,000 Btu
Heating effect coefficient	0.7
Wall and glass loss for convection heater	77,700 Btu
Add infiltration loss	29,000 Btu
	106,700 Btu

By test the average Btu input to this residence on a zero day was 110,000 Btu.

Conclusions

1. That the heat loss from any given room or building is dependent upon the source of heat supply.
2. That *heating effect* measures the change of the overall coefficient of heat transmission for a wall under the influence of different sources of heat.
3. That the *heating effect* coefficient applies only to wall and glass loss and not to infiltration loss.
4. That convection heaters should be rated in Btu output at the outlet grille or opening.

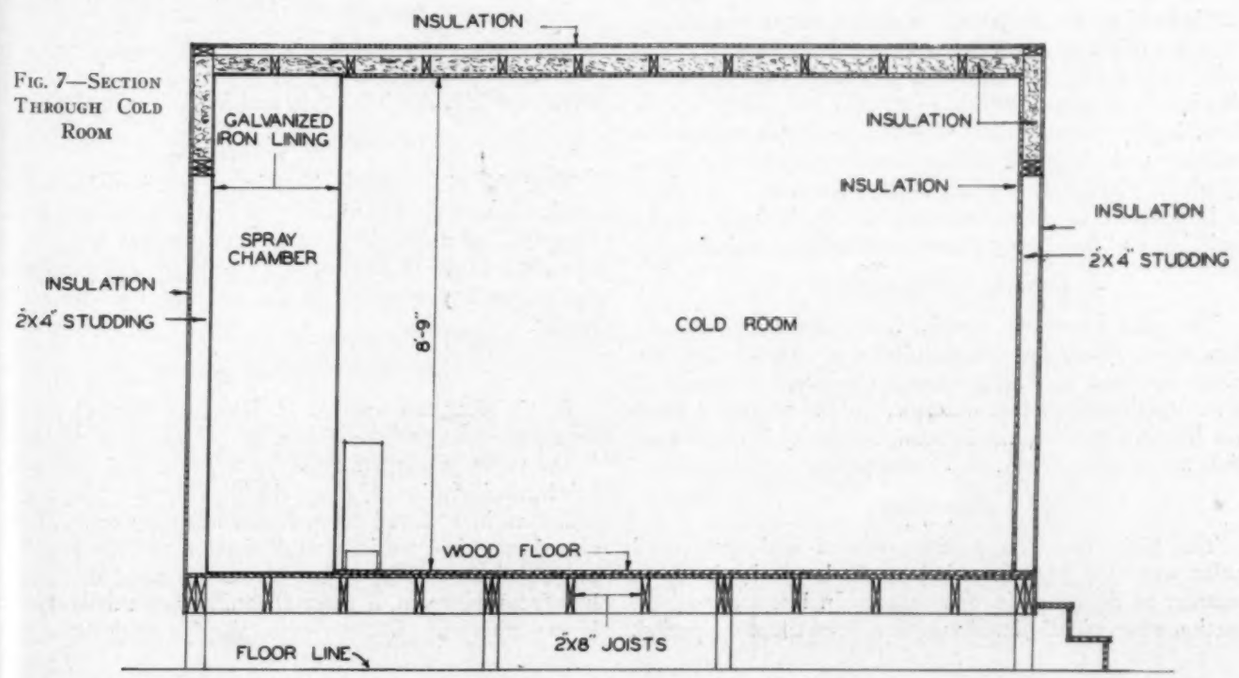


FIG. 7—SECTION THROUGH COLD ROOM

ADDENDA

After completion of the data recorded in this paper the following tests were conceived for the purpose of furnishing additional proof of the existence of radiant energy losses.

So far as the heating of the air in a room is concerned, it has been proved that heaters of different types can accomplish the same work with different heat inputs. In other words, it is possible to maintain the same floor-line and breathing-line temperatures with a heater condensing less steam than some other heater.

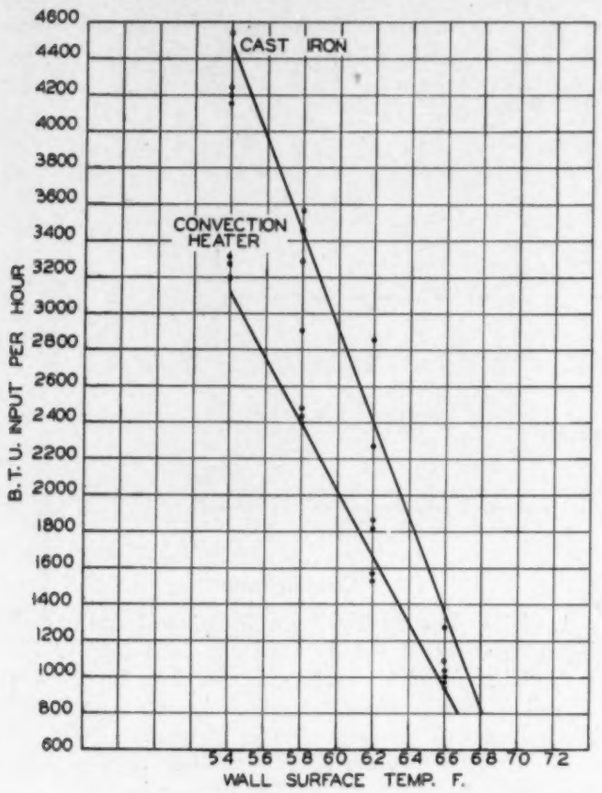


FIG. 8—RESULTS OF TESTS OF EXPOSED AND ENCLOSED HEATERS

It is a generally accepted fact that radiant energy emission is a function of surface roughness and color. Therefore, highly polished surfaces are poor radiant energy emitters and also poor radiant energy absorbers. Thus, by nickel plating the surface of the cast-iron "radiator," it is possible to eliminate most of the radiant energy emission and leave only the convection heat emission.

Laboratory Equipment

The cold room, as described and illustrated in this paper, was used for determining the capacities of the plain cast-iron and nickel-plated sections. In addition, a set-up similar to that described in the proposed Code for Testing and Rating Concealed Gravity Type Radiation was used for comparative purposes.

Procedure

The plain cast-iron section painted with gray lead paint was tested for a period of six hours in the same manner as described in this paper. In like manner, the section when nickel plated was tested for the same period. The hot room still air tests were run for a period of two hours each. During the cold room tests, the same breathing-line temperature—namely, 72 F—was maintained with both the nickel plated "radiator" and the painted "radiator" section. The same standard of heating perfection existed in both cases.

Test Results

The results of the tests are as follows:

Cold Room Test. A gray painted cast-iron "radiator" section condensed 1.47 lb of steam per hour. Same section (nickel-plated) condensed 0.71 lb of steam per hour. (*Note*: Wall temperature 66 F in both cases. Breathing-line temperature 72 F in both cases.)

Hot Room Test. Cast-iron section (painted) condensed 1.50 lb of steam per hour under standard-testing conditions.

The nickel-plated section condensed 0.75 lb of steam per hour.

Discussion of Results

It is to be noticed that the results obtained in the cold room checked very closely with the results obtained in the hot room. It seems apparent from this that a measure of the output of a "radiator" can be obtained accurately by the hot-room method as provided for in the proposed code. This is also one of the conclusions given in this paper.

The ratio of condensation of the nickel-plated "radiator" (one section) to that of the painted "radiator" is 50 per cent. This factor is lower than that determined for the convection heater. This discrepancy between these two factors probably lies in the fact that the "radiators" originally tested were of more than one section, and, consequently, emitted a smaller percentage of radiant energy per square foot of rated surface.

The purpose of the test was to demonstrate by means of a known factor—namely, that a polished surface reduces the radiant energy emission—that the change in the overall coefficient of transmission of a wall (U) when convection heaters are used is due to the elimination of radiant energy effects.

The results obtained check very closely those reported in Bulletin 120, University of Illinois Experiment Station, page 115, for bright tin and black iron cylinders.

Conclusions

That the difference in heat input required by a convection heater (nickel-plated cast-iron or non-ferrous) as compared to a cast-iron "radiator" to heat a room to equal standards of perfection is due to the elimination of radiant energy emission in the case of the convection heater.

DISCUSSION

A. C. WILLARD AND A. P. KRATZ (WRITTEN): Before proceeding to the discussion of the data contained in the paper presented by Messrs. Trane and Scanlan, a brief consideration of the principles involved in testing radiators in a closed room is not out of place. Such a closed room is essentially a calorimeter, and the total heat output of the radiator must be exactly equal to the heat loss from the room, if thermal equilibrium exists. Hence, if two different radiators maintain the same mean tem-

perature for the air in the room, and exhibit a difference in steam condensation, or heat output from the radiator, the conclusion is inescapable that the heat loss from the room must be different in the two cases.

The change in heat loss from the room may result from a redistribution of temperatures at the various levels or from a change in the inside surface coefficient of heat transmission, as suggested in the paper under discussion, but in any case the change must be compensated for, either by a difference in size, or efficiency of heat transmission in the radiators, in order to produce the required difference in steam condensation. The difference in steam condensation directly reflects the change in heat loss from the room. Therefore, comparisons made on the basis of steam condensations obtained in a room in which the changes in heat losses under the given circumstances deviate widely from those that would occur in a room with practical standard construction under the same conditions, while interesting as applying directly to the room in which the tests were made, have no interest nor significance as applied to the actual rooms of standard wall construction in which the radiators must later be used.

It is obvious from the tests made by Messrs. Trane and Scanlan on radiators in a room with water cooled walls, that, at the same breathing level temperature for a given air to wall surface difference in temperature, there is a change in the *inside surface coefficients* as shown and computed in the paper.

Thus, the tests show:

$$f_{c.t.} = \frac{3440}{166\ (71-58)} = 1.59$$ Btu for a metal wall, heated by a bare cast-iron radiator.

$$f_c = \frac{2400}{166\ (71-58)} = 1.11$$ Btu for a metal wall, heated by a convector.

The heat lost from the room is determined by the overall coefficient of heat transmission, U, the complete expression for which is:

$$U = \frac{1}{\frac{1}{f_1} + \frac{1}{c} + \frac{1}{f_2}}$$

in which $\frac{1}{f_1}$ = resistance of inner surface of wall, including radiation and convection, and also conduction of air film.

$\frac{1}{c}$ = resistance of the wall itself, where c = conductance.

$\frac{1}{f_2}$ = resistance at the outer surface of the wall, involving the conductance of the air or water film.

In the case of the water cooled wall, f_2 is of the magnitude of 350[1] Btu per square foot per hour per degree Fahrenheit, thus making $\frac{1}{f_2}$ of the order of 0.00285. The thickness of the metal wall was not given, but it is hardly greater than ⅛ in. The conductivity, k, for sheet steel is of the magnitude of 400 Btu per square foot per hour per degree Fahrenheit per inch. Using these values, $\frac{1}{c} = \frac{0.125}{400} = 0.0003$. *Hence,* in the case of the water cooled wall both $\frac{1}{c}$ and $\frac{1}{f_2}$ could be safely neglected. However, including them for the sake of completeness, the overall heat transmission for the water cooled walls used in connection with the two radiators becomes

$$U_{c.t.} = \frac{1}{\frac{1}{1.59} + 0.00285 + 0.0003} = \frac{1}{0.63215} = 1.582$$

$$U_c = \frac{1}{\frac{1}{1.11} + 0.00285 + 0.0003} = \frac{1}{0.90415} = 1.106$$

$\frac{1.106}{1.582} = 0.699$, or 30.1 per cent decrease.

Using *precisely these same values of inside surface coefficients* for an actual wall, such as those of the room heating testing plant in the University of Illinois, the overall heat transmission values are:

$$U_{c.t.} = \frac{1}{\frac{1}{1.59} + \frac{1}{0.30} + \frac{1}{2.6}} = \frac{1}{4.34} = 0.230$$ Btu for an actual wall heated by a bare cast-iron radiator.

$$U_c = \frac{1}{\frac{1}{1.11} + \frac{1}{0.30} + \frac{1}{2.6}} = \frac{1}{4.62} = 0.216$$ Btu for an actual wall heated by a convector.

$\frac{0.216}{0.230} = 0.940$, or a decrease of 6.0 per cent.

The value 0.30 is the conductance, and the value 2.6 is the outside surface coefficient for the actual wall in the test plant at the University of Illinois, when the air to inside surface temperature difference is (71 — 58) or 13 F, which is the condition on which the Trane water cooled wall test plant was operated in the tests of the two different radiators.

Finally, since the decreases of 30.1 per cent in the Trane plant and 6.0 per cent in the University of Illinois plant in the overall heat transmission coefficients, when convectors are used in place of exposed cast-iron radiators in each plant, are almost exactly equal to the decreases in steam condensation in the two plants respectively, it is apparent that a relatively larger exposed cast-iron radiator had to be used in the plant with water cooled walls to maintain the same breathing level temperature and inside wall surface temperature in both plants.

This shows the fallacy of using water cooled walls *where the sizes of the radiators are adjusted to maintain a fixed temperature at the breathing level,* and then drawing conclusions based on condensation alone without taking account of the adjustments in radiator size

[1] Heat Transfer in Ammonia Condensers, Part III, Univ. of Ill. Eng. Exp. Sta. Bulletin 209, Fig. 19, page 35.

that must be made with rooms having walls with different heat resistances.

It is obvious that a change in the inside surface coefficient takes place with different radiators, but conclusions based on tests in a given room will be valid only when the ratio of surface resistance to total wall resistance is comparable with that found in actual service walls.

R. V. Frost (Written): The paper by Messrs. Trane and Scanlan is a splendid contribution to the current research on heating effect.

In the choice of laboratory equipment it is to be noted that for this work there has been selected a type of room that permits the isolation and separate study of the various factors bearing upon heat transfer. In the writer's experience this method is much more effective than to attempt to study the causes for heating effect by the aid of a test room built to simulate actual structural practice. There is no sound reason why in our research upon effective heat we should stick to test equipment that permits nothing other than general conclusions when by the proper selection of equipment, studies can be made of specific characteristics which permit one to accurately analyze each individual heat transfer factor that enters into the phenomenon of space heating.

To do this it is not necessary to build a room that is inflexible in heat loss characteristics. A test room can be as easily built to permit tests with different wall constructions as to confine the tests to a single type of construction. Criticisms have been frequent against the water cooled wall type of test room but if those making such criticisms will take the trouble to weigh all the factors, the reasons for their objections will be found insignificant in the face of those in favor of the water cooled walls. In the conduct of radiator tests, the most severe heating condition must be reproduced under controlled conditions. Obviously the most severe heating condition is that of a glass enclosed room exposed to a cold driving rain.

A typical room of this type is illustrated. A water cooled wall test room reproduces such a condition. Radiation must be rated to take care of the worst condition. Any heating load less than the worst need not concern us unless we are making a specific study of wall construction, for if a radiator will suffice for the worst weather condition it will of course prove adequate for any other. Study of wall construction must not be confused with the separate study of the radiator. They are related but not interdependent.

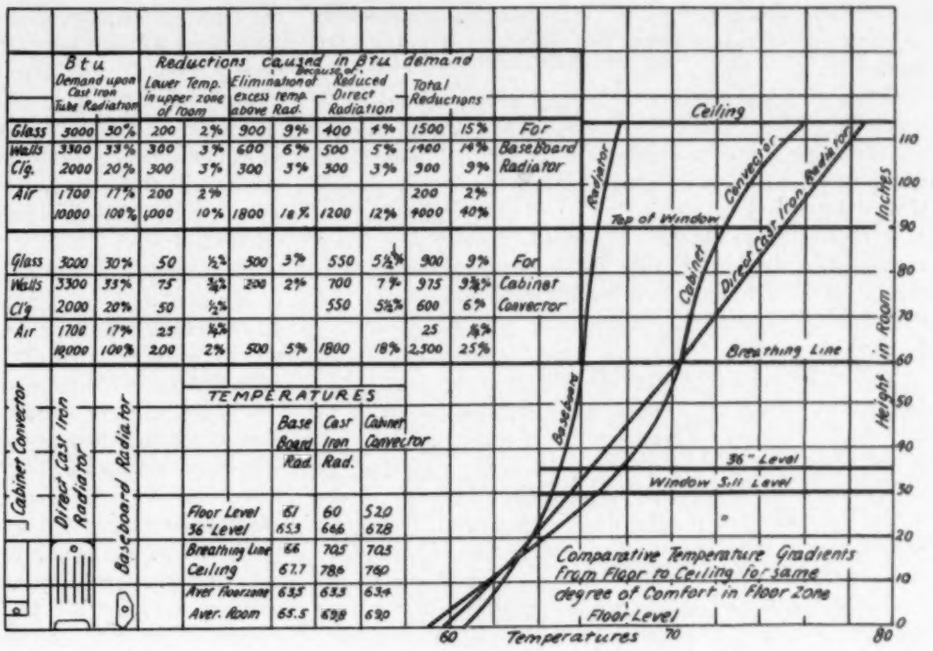
Messrs. Trane and Scanlan have taken a step in the right direction by the elimination of infiltration in this study. As they continue they will find it essential to more specifically study other factors. However while infiltration as a factor has been eliminated, it is not to be ignored and requires a special investigation.

One criticism to be made of the Trane experiments is that too few measurements of temperature have been taken. If temperature readings had been taken in at least thirty more locations in the room, data could have been gathered that would give a very much more complete picture. In the writer's experimental work on heating effect he has found it quite essential to base the investigation upon a study of the floor to ceiling temperature gradient. When this is done and account is taken of the temperature range throughout the room the study of heating effect is quite simplified.

In the Trane tests a uniform temperature of 72 deg at breathing line was selected. Had the temperature gradient been studied instead, the uniform breathing line temperature would not have appeared so desirable. Prof. Willard has proposed that a uniform temperature at the 30-in. level be used in comparing radiators. This is a step in the right direction, but as a result of the tests in the writer's laboratory the average temperature in the zone from floor to 36 in. above floor has been demonstrated as the best for comparison. This zone, which has been designated as the floor zone, is the one occupied by a person in a sitting position—the shoulder height being

Typical Room With Large Glass Exposure

HEATING EFFECT DETERMINATIONS IN A TYPICAL ROOM REQUIRING 10,000 BTU OF HEAT DEMAND IN ZERO WEATHER AND 15-MILE WIND



at about the 36-in. level. This zone must be maintained at a comfortable temperature if the occupants of the room are to enjoy genuine heating comfort. A person standing does not require as much heat as one sitting, so that to judge the heating of a room by the comfort of a person in a standing position is far from correct. Of two radiators, each giving 72 deg at breathing line, one may cause a very much cooler temperature in the zone below the 36-in. level and thus be a less desirable radiator from the comfort standpoint. (See chart with temperature gradients of three different types of radiators.) On the other hand, of two radiators providing the same degree of comfort in the floor zone one may produce a temperature of say 4 deg less at the breathing line. To compare radiators by the breathing line temperature only is a practice that cannot, in the light of our present knowledge, be permitted to continue.

Likewise, to compare radiators strictly on the basis of temperature at the 30-in. level is incorrect, for some radiators particularly of the cabinet convector type with horizontal outlet, will give a high temperature at the 30-in. level but a sub-normal temperature at the floor level. The temperature gradient chart of the three different types of radiators shows how comparison on any one level is incorrect.

In the Trane tests the assumption has been made that heating effect is solely a function of radiation. Here too a study of the temperature gradients shows that such an assumption is not correct. Both convection and conduction are important factors and since a poor radiator is also both a poor conductor and a poor convector, it is quite easy to confuse the effect of one with that of the other. Much of the heat loss that we have attributed to radiation the writer has found instead to be caused by direct conduction from the air to the glass or wall surface, for much of the air coming in contact with the glass and wall surface does so solely through convection.

Messrs. Trane and Scanlan have chosen to designate the heating effect coefficient as a decimal, such as 0.7 or 0.85 etc. In the writer's work the complement of the decimal has been used, namely, 0.30 or 0.15, and the term has been expressed as a percentage, 30 per cent or 15 per cent, etc.

There are many reasons why such a designation is more practical, a few of which can be named.

1. Those accustomed to speaking of heating effect have formed the habit of saying one radiator has a better or poorer heating effect than another. Such an expression implies a higher or lower heating effect so that it is natural that a high heating effect should be designated as, say 40 per cent, while a low heating effect might be 10 per cent. Since a cast-iron radiator of say 38 in. 5-tube is regarded as unity then a radiator of 10 per cent heating effect uses 10 per cent less heat input than the cast-iron radiator, while a radiator of high heating effect may have 40 per cent heating effect and use 40 per cent less heat input.

2. In view of the above, the saving in heat input means the same percentage in fuel saving, which when floor zone comfort is taken as the index, is the sought for result of heating effect. Thus a radiator of no heating effect percentage gives no fuel saving but a radiator of 40 per cent heating effect will give 40 per cent saving.

For these reasons the percentage method of designation as outlined above appears to be the more logical and more readily understood by the lay mind.

A further criticism of the Trane-Scanlan paper is the plotting of the chart, Fig. 8. In the plotting for the convection heater the line drawn does not pass through the average of the points plotted. This results in an erroneously favorable comparison for the convection heater and further implies a relationship between wall surface temperature and Btu input that cannot be true. From the line as plotted, using the Trane method of expressing the coefficient, we find this to be 69 deg when the wall temperature is 58, while at 66 deg wall temperature it is still 69. A more correct location of the line and one in keeping with the findings of others gives coefficients of 73 and 78 respectively. Obviously as the temperature of the exterior of the wall approaches that of the room the heating effect ratio approaches zero and the cast-iron radiator would become fully as effective

as the convection type heater. Therefore, the comparison as plotted by Messrs. Trane and Scanlan cannot be correct.

For a practical method of indicating heating effect in numerical values the writer worked out a plan shown on the chart here presented. The method admits of ready calculation and removes much of the mystery with which the theory of heating effect is now regarded.

It is not clear whether Professor Rowley had some such plan in mind in offering his paper on, Essential Elements for Determining Heating Plant Requirements. At any rate his arguments give plenty of support to the method proposed.

It is to be seen from the chart that in the calculation for heating effect the heat loss calculations are made in the usual manner. This demand is taken as that imposed upon a standard cast-iron radiator, then for the radiator or heater whose heating effect is to be determined, a temperature gradient is prepared from test data gathered from comparison with a standard cast-iron radiator. The temperature gradients are laid out on the same chart and so located that temperatures in the floor zone are the same average. From these gradients, temperatures can be determined for wall surface, glass, ceiling and infiltration. Since from a cast-iron radiator the largest proportion of heat emitted streams directly upward over the window surface, the wall above, and across the ceiling, account must be taken of the higher surface temperatures over this area. These items are separately calculated and then compared with the known heat demand difference between the radiator in question and the standard radiator. The sum of the separate items is found to be less than the difference between the two radiators and this balance can be attributed to direct radiation. Thus the effect of direct radiation is determined by difference rather than by calculation. There seems to be no method at present known by which the effect of direct radiation can be exactly determined. This is a problem now under investigation in the writer's laboratory.

With the advance that has been made in the study of heating effect in the past few months there is good reason to expect that very shortly all types of space heaters will be graded according to their heating effect. The baseboard type of radiation shown on the chart has been under observation in the writer's laboratory for the past four years and is now put out on a strictly heating effect rating by evaluating the rating according to the plan here shown. As a result of this method of presentation all mystery has been removed and the explanation of its heating effect accepted as entirely logical, although it has been found possible as can be seen from the chart to give it a heating effect rating as high as 40 per cent—a value that is truly staggering, when it is realized that this percentage also represents the actual saving in fuel over that obtained with a standard cast-iron radiator when the radiators are operating under zero weather conditions.

E. J. Vermere, (Written): Without question this paper is an important contribution to the progress of heating and in the future years will be considered as a milestone marking a definite step toward better engineering. For several years we have discussd heating effect, although we have not always used this term and many times where a fuel saving may have occurred due to heating effect, credit for this economy has been given other equipment in the system and not to the type of radiator used. This is the first example we have had where heating effect has been discussed beyond the theory stage and where a definite conclusion has even been attempted. If subsequent experiments bear out the conclusions reached in this paper, it will certainly prove that much of our highly advertised engineering ability is still based on thumb rules.

Most engineers will admit that in commercial practice it is impossible to obtain heat loss calculations within 25 per cent of actual requirements and that such a margin of error is certainly a heavy tax on the man who pays for the installation.

I would like the authors to give more of the information which led them to make the two statements in the second paragraph on page 786, *first*: "No material difference could be determined between heating effect coefficients for high and low radiators. If the results of the tests for the cast-iron radiators as given in Table 1 are plotted separately instead of being given as the average results, it will be found that the 5-tube 26 differs materially from the 5-tube 38 and the wall radiator. A wider range of similar tests would answer this question." *Secondly*: The authors state, "likewise, no appreciable difference could be detected between a convection heater with a high stack and one with a low stack." It seems reasonable to believe that this would be true for various heights of convection heaters of the same type where the outlet grille is below the breathing level, however, the question arises whether tests have been made with high outlets, especially where the outlet is near the ceiling line. If such tests have been made then our theory of air stratification which prevents proper heat diffusion with a high outlet must be incorrect. Actual experience with installations where outlet grilles are at or near the ceiling have indicated improper heat distribution and therefore the same heating effect factor could not apply.

Another basic theory of heat diffusion which we have accepted without much supporting data is ignored in these conclusions; that is the relation between heat diffusion and temperature of the air leaving the outlet. For all practical purposes, the unit heater is to be considered a convection heater. We also know that high outlet temperature units will not give as good diffusion as low outlet temperature units of equal capacity. On page 135 of The Guide 1931 it is recommended that the delivery temperature generally should not be more than 70 deg above room temperature desired. Therefore it seems reasonable to believe that outlet temperature will have a decided bearing on the heating effect factor.

The experience of the last ten years in the application of unit heaters is conclusive evidence that the factor of heating is primarily a relation of convected heat. Many manufacturers of this type of equipment claim fuel savings of at least 25 per cent and in most cases this economy can only be attributed to the heating effect factors due to convected heat, thus supporting the conclusions of the authors.

This entire subject is of such great importance to everyone connected with the heating profession that I feel it would be in order for the Society to take immediate steps to either approve or disprove these claims by ordering our director of the research laboratory to

begin at once the necessary experimental work incident to a thorough analysis of this subject. It is easy to forecast the controversies that will arise on this subject between advocates of different types of space heaters and our Society is the one agency which can analyze this subject in an unbiased manner and whose findings will be accepted by the profession.

N. D. Adams (Written): During some experiments run at La Crosse in 1926 to determine the adaptability of cast-iron radiation to stack heating, it was noticeable that the condensation required to maintain a constant room temperature was less than that required by direct radiation, that is, exposed cast-iron radiator.

We could not understand at that time why this was so, and as we were interested in the problem of whether or not the cast-iron radiator was as efficient a convector as the so-called copper radiation, we did not look further for the cause.

The results of our experiments at that time indicated that the cast-iron radiator has approximately the same efficiency as the copper radiation when used as a convection heater; but due to the decreased weight of the copper radiation, ease with which it could be housed, and the fact that it gave less resistance to the flow of air than the cast iron, we decided to specify it for the particular job at hand. This installation has been in operation for three years, proving very satisfactory.

Quoting from Allison J. Thompson in Man, Nature and Civilization: "The so-called chaotic condition of nature is largely a conception of the human mind—for we may be sure that all of the various laws, forces and processes of nature are functioning with respect to each other, each in its own peculiar manner, with what may be regarded as an uncanny precision, wherever and whenever they come into conjunction with each other. The more we learn about the great scheme of nature and the inter-relation of its various elements the less of chaos is found."

To understand the inter-relationship of these various elements of nature and the various laws, forces, and processes by which they are governed is the task of our research work. We are to be congratulated that there are men among the members of this Society who are willing to give their personal time and learning, backed by the financial resources of the industries which they represent, to the end that we may give to the future generations facts concerning the operation of the forces of nature, which will increase their comfort and decrease waste effort which so many of us expend in combating, working in opposition to these forces. This paper shows a thoroughness of preparation, untiring effort to exclude errors, and conclusions which we believe are fundamental and have a practical application to the industry as a whole.

In discussing this paper it seems to me that there are three questions which should be asked. (1) What was the incentive force which caused these experiments to be made? (2) Were the conditions surrounding the experiments such that the conclusions may be taken as facts? (3) If these conclusions are facts, then how may we apply them to the practice of heating and ventilating?

Some of the incentives which have served as stimuli for reactions in human and animal life are ambition, pride, good-will, philanthropy, curiosity, and the fascination of adventure, exploration, and research, together with the glory of accomplishment.

Much of the research work which has been done by individual members of this organization has as its incentive, the desire of improvement of manufactured products, in an effort to gain leadership in some division of the industry. The men who have spent their time so generously have been rewarded with the glory of accomplishment.

Another incentive is that there has been no time in the history of business enterprise when it was so necessary to conserve the natural resources and to eliminate waste, thus decreasing operating overhead.

As for the conditions under which the experiments were made, let us consider the cold room which was built in La Crosse in which these experiments were conducted. This particular laboratory was built, after spending considerable time in studying the results obtained from a predecessor in the University of Illinois. As we discuss the arrangement of this particular cold room let us keep in mind that Mr. Trane and Mr. Scanlan sought to determine the cause of one effect in particular and therefore wished to exclude any conditions which previous experiments had shown conclusively did not affect the problem at hand.

As we study the details of their room, we notice the omission of windows in the exposed wall, the absence of attic space or basement, and in particular, the construction of an exposed wall surface which is of sheet metal definitely eliminating the possibility of infiltration.

We read in the University of Illinois Bulletin No. 31, Investigation of Various Factors Affecting the Heating of Rooms with Direct Steam Radiators by A. C. Willard, A. P. Kratz, M. K. Fahnestock, and S. Konzo, several interesting conclusions.

Conclusion 18. The use of storm sash alone does not materially affect the room air temperature conditions, but did effect a gain in steam economy of approximately 11 per cent in the particular room tested.

Conclusion 19. The use of a storm door alone improves room air temperature conditions, and in the particular room tested effected a gain in steam economy of approximately 21 per cent.

Conclusion 20. The use of both storm door and storm sash improves room air temperature conditions and in the particular room tested effected a gain in steam economy of 31.7 per cent. The actual saving in general will be dependent on the ratio of window and door area to exposed wall area, and on the air-tightness of the storm door and sash.

It is hardly necessary to discuss further that the omission of windows in exposed wall surface, for this particular problem only eliminated one of the variables which they would have to allow for in each test. We now believe that there is a direct relationship between the economy of a heating system and the amount of infiltrated air; a constant based on the temperature differential.

Conclusion 21. Variations in basement and attic temperatures do not have a material effect on room air temperature conditions.

Looking at the cross-section of the LaCrosse cold room, we note that the ceiling and front wall were well insulated while the floor was of a regular house construction, double flooring, open below to the room tem-

perature. The paper does not state whether or not a constant temperature was maintained in the space surrounding the cold room. We believe, however, that such was the case and that a constant transmission was maintained during the eight hours of each test between the chamber and the surrounding space. If the outside room conditions were maintained constant for all types of radiation tested, then the elimination of attic, basement and a front wall construction can have no effect on the results obtained.

The paper states that the sheet metal wall construction was used in order to eliminate infiltration and thus simplify the problem. Is it necessary that the coefficient of transmission of the exposed walls (U) be the same for two different experiments? No two buildings have exactly the same exposed wall construction. The important factor is that, before starting tests, the coefficient of transmission of the chosen wall construction be definitely established. This we believe was done in this case, as 38 tests of eight hours duration each were conducted to determine the operating peculiarities and characteristics which would affect the operation of the cold room.

We note that during the tests, for each eight-hour period, the inside wall surface was maintained at a constant temperature. The temperatures used were those found in the experiments conducted by Prof. A. C. Willard and Prof. A. P. Kratz at Urbana.

Conclusion 24. The temperature of the inside surface of exposed standard frame walls varies from 67 to 59 F for walls not exposed to air movement, and from 65 to 53 F for walls exposed to an air movement of approximately 10 mph as the outdoor temperature ranges from 40 to —10 F and the indoor temperature remains constant at 72.5 F.

It is interesting to note that the overall heat transmission of this construction approximates that of the standard wall used at Urbana; when the average indoor temperature was maintained at 71 F with a —10 deg outside temperature. There was a heat loss of 20 Btu per square foot per hour.

We have referred to the 38 tests of eight-hour duration which were conducted to determine the proper procedure for operating the cold room during the standard tests.

In all experimental work there is a question of proper methods in obtaining data, number of tests taken, and duration periods of tests in order to eliminate possibilities of error that might be due to fluctuating conditions. The consistency of the results as shown in Table 1 would indicate that sufficient number of tests, time and care had been taken.

Should some aviator attempt to establish a new altitude record, standard instruments are required and conditions imposed as set down by a committee of the *American Aviation Society* at the Bureau of Standards, Washington, which must be followed in order that the aviator may receive credit for his efforts if successful.

Could we, as a Society, through our Research Committee, set up a code of standard rules and regulations to be followed. Then in the event that these rules and regulations did not apply to new work, the committee could be notified before the test was to be made and have a representative on hand to approve the conditions and procedure, and certify the results.

If we now accept the conclusions of this paper as facts, let us consider how they may be applied to the practice of heating. The illustration given on page 787 for the cold room at Urbana shows a heat loss of 140,000 Btu, while the Btu required if a convection heater is used is stated as 106,700 or 76 per cent of the calculated heat loss. This makes a savings of 24 per cent in steam consumption. If this be true, then, the radiation of any steam heating system to be installed hereafter should either be nickel plated or otherwise treated to prevent the giving off of radiant energy or some type of convection heater should be used.

It is also possible that there are many buildings using cast-iron radiators today where it would pay the owners to investigate the possibilities of replacing the cast-iron radiators with convection heaters, or enclosing the present radiators to convert them into convectors, or to nickel plate them, in which case the surface may not be sufficient for the space requirements.

Example

Total square feet of radiation in hotel building	30,000
Season heat demand per 100 sq ft	53,000 lb
Total season demand for the building	15,900,000 lb
Total season's cost at $1.00 per thousand pounds	$15,900
Total cost to replace building with cast-iron radiation with a non-radiant type at 50 cents a square foot	$15,000
24 per cent saving in steam demand	$3,816
6 per cent interest on owner's investment	$900
Net saving per year	$2,916

In addition to the economy created by the use of the convection heaters, we cannot fail to notice the increase of comfort in the room due to the smaller differential between the floor and ceiling temperature.

H. L. Stiles (written): Their conclusions verify certain experiments we also have made relative to electric air heating radiation. Although we have not had the advantage of conducting our tests in a room in which the temperature conditions could be so closely controlled, the test results indicate a trend towards those secured by Trane and Scanlan.

The test room in which we worked measured 15 ft x 12 ft x 9 ft, with two 3 ft x 5 ft windows in an east wall. The door, 2½ ft x 7 ft, was located in the north wall. The wall and ceiling were insulated with approximately 4 in. of rock wool, while the floor was built up of 6 in. of concrete, sleepers, tar paper and flooring.

Preliminary to our tests we were of the opinion that the radiant type of electric heater was suitable, where an occasional, localized heat is required, while the purely convection type heater was intended primarily for continuous heating where the air was to be maintained at some predetermined temperature. There is, however, a deficiency of literature on this subject, so, therefore, we made our own investigation in this field.

The radiant type of heater used in our tests operates with electric coils at a bright red temperature and from previous radiometer tests showed a radiation efficiency of 37 per cent. It seemed apparent that this type of heater would be glaringly inefficient, since 37 per cent of its heat would be distributed over cold walls, doors, and windows to be lost by transmission.

The convection heater, on the other hand, since it showed 97 per cent convection efficiency, seemed to be the efficient heater for continuous heating since the radiation losses are absent and the air is continuously cir-

culated and recirculated by natural draft through the *convector* and heated. Also the units, as contrasted with the radiant heater, operated at a *black heat*.

In the table below some of the data from the convection and radiant heater tests are tabulated, as well as the results obtained when a 14-in. fan was used to circulate the air in the room while the convection heater was tested.

Type	Watt-hrs. used per cu ft per degree rise (Raising temp.)	Temp. Grad. Floor to Ceiling	Watts input per cu ft per degree diff. between inside and outside Maintaining Temp.
Convection Heater....	0.074	18½	0.020
Convection Heater with Fan	0.066	7	0.036
Radiant Heater	0.1203	11.5	0.019

The watt hours used per cubic foot per degree rise was the average value secured by dividing the total watt hour input to raise the temperature to 70 deg at the 5-ft level, by the temperature rise and the cubical volume of the room. The temperature gradient was taken when the thermometer at the 5-ft level was 70 deg. The watts input per cubic foot per degree difference was also an average value obtained by dividing the total watt hours consumed during the period of the test by time, temperature difference between inside and outside, and the cubical volume of the room. The outside temperature was in all tests between 30 and 40 F. Both heaters were rated at 3,000 watts each.

The first test, on the convection heater, might be taken as a standard for comparison. The temperature gradient between floor and ceiling for this heater was 18.5 deg, which is about average for a 9-ft ceiling.

It will be noted that by circulating the air in the room, the watt hours used per cubic foot, etc., were decreased as shown in test No. 2 when compared with test No. 1. This would indicate that, since the air was continually stirred up, the heat was not allowed to pocket at the ceiling, but was utilized in the living zone. Further, proof of this was evident from the decrease in temperature gradient from 18.5 deg to 7 deg. On the other hand, the watts input, etc., was increased materially, indicating that the air movement over walls, ceiling and floor tends to increase losses to these areas.

The radiant heater in test No. 3 shows that during the heating up period the watt hours consumed was greater, due no doubt to the absorption of the radiant heat by the walls, ceiling, furniture, etc. This absorbed heat is to a large extent recovered later, although there is considerable wall transmission loss. No attempt was made to determine this loss quantitatively but it is possible that, though it may not be excessive, it is appreciable as later discussed.

It will be also noted that the radiant heater watts input per cubic foot per degree difference between outside and inside temperature, which was maintained at 70 deg at the 5-ft level, was slightly lower than for the convection heater. This condition seemed improbable, but repeated tests gave verifying data. When it is considered that this radiant heater had 37 per cent radiation efficiency, these results are of significant importance. Our analysis of the data is that the heat radiation from the radiant heater falls on all exposed surfaces in the room, heat-soaking them until a condition of equilibrium is reached, when they in turn become effective as radiators and convectors. This extremely large area of low temperature heating area induces but a small temperature rise in the surrounding air. This is an ideal condition for heating since it results in a low temperature gradient between floor and ceiling.

Thus it seems that the radiant heat received and absorbed by the walls, etc., is partially lost to the outside, while the remaining portion is given up to heat the air in the room. The part given back into the room is utilized sufficiently efficiently as to more than compensate for the loss and results, therefore, in a condition better, or at least equivalent to the convection heater.

What Gives Air Its Stimulating Quality?

Research Reveals Amazing Results

Read the October Issue

Changes in Ionic Content of Air in Occupied Rooms Ventilated by Natural and by Mechanical Methods

By C. P. Yaglou (*Member*), L. Claribel Benjamin and Sarah P. Choate (*Non-Members*)

This paper reports the results of a year's intensive investigation at the Harvard School of Public Health made in co-operation with the A. S. H. V. E. Research Laboratory and deals with fundamental changes in the ionic condition of the air in occupied rooms ventilated by natural and by mechanical methods and with the influence of various air conditioning processes upon the ionic content of the air. Since the field is comparatively new, a brief discussion of the basic principles of atmospheric ionization is also included.

Cleveland

January 25-29, 1932

DURING the week of January 25 to 29, 1932, Cleveland will be the gathering place for engineers interested in heating, ventilation and refrigeration, because three events of national importance will be held simultaneously: the 38th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, the 27th Annual Meeting of the *American Society of Refrigerating Engineers* and the Second International Heating and Ventilating Exposition.

The central location of Cleveland, the attractive technical programs and the display of new equipment by manufacturers will be the magnets drawing the crowd to Cleveland and it is expected that this will be the most notable week in the history of the profession and industry.

Annual Meeting headquarters of the A. S. H. V. E. will be the Hotel Statler, and sessions of the *A. S. R. E.* will be at the Hotel Cleveland, while the Heating and Ventilating Exposition will occupy the Cleveland Auditorium. Arrangements have been made for a joint session of the A. S. H. V. E. and *A. S. R. E.* and subjects of mutual interest will be discussed.

The Cleveland Chapter will be hosts to the A. S. H. V. E. and activity has already started under the direction of T. A. Weager, General Chairman of Arrangements. The Committees already appointed are:

Finance Committee—Walter Klie, *Chairman;* W. E. Stark.

Banquet Committee—W. C. Kammerer, *Chairman;* C. W. St. Clair, Vincent Eaton.

Reception and Registration Committee—C. F. Eveleth, *Chairman;* F. A. Kitchen, C. Gottwald.

Transportation and Inspection Trips—F. H. Morris, *Chairman;* C. J. Deex, J. E. Beyer.

Entertainment Committee—R. G. Davis, *Chairman;* C. W. Colby, E. H. Pogalies.

Publicity Committee—M. F. Rather, *Chairman;* D. E. Humphrey, Akron District.

An extensive technical program is being prepared and it covers a comprehensive list of subjects. The first technical papers appear in this issue, Conductivity of Concrete by F. C. Houghten and Carl Gutberlet and Study of the Combustible Nature of Solid Fuels by R. V. Frost. In addition to the joint technical sessions the local committees of the A. S. H. V. E. and *A. S. R. E.* are planning some cooperative social events for the entertainment of members of both organizations.

Ample time has been reserved for attendance at the Second International Heating and Ventilating Exposition and with the progress now made it is evident how diversified the interests of the industry are. Two sections, one reserved for refrigerating equipment and the other for air conditioning apparatus, will present many new designs which will be displayed for the first time. Other new equipment on display will be found in the sections devoted to unit heaters, gas burning appliances, oil burners, stokers, boilers, furnaces, fans, pumps and specialties.

The announcement by Charles F. Roth, manager of the Exposition, indicates the selection of an attractive poster seal for the use of exhibitors. This seal gives the impression of heat and air movement through the use of a conventionalized flame and fan wheel appearing against a gray background. This special seal will soon be carrying the message of the Exposition to every part of the country.

A record attendance is anticipated and plans to join the crowd at Cleveland should be made now.

CLEVELAND AUDITORIUM

A Survey of the Society's Research Interests

WITH the purpose of rendering a practical and worthwhile service to the members of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEER, as well as to the heating and ventilating profession and industry at large, the Society's Committee on Research recently mailed the following letter to Society members and others, and invited attention to a reprint of the announcement published on page 519 of HEATING, PIPING AND AIR CONDITIONING, June, 1931.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

51 MADISON AVENUE
NEW YORK, N.Y.
TELEPHONE ASHLAND 4-0291

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F. C. MANTON
W. A. ROWE

June 20, 1931

Mr. W. H. Carrier, President
American Society of Heating and Ventilating Engrs.
850 Frelinghuysen Ave.
Newark, N. J.

My dear Mr. Carrier:

The Society's Committee on Research desires to ascertain what problems of a strictly engineering type would make profitable subjects for technical study or investigation by our Research Laboratory or through our affiliations with Universities or Government Bureaus.

You, no doubt, have need of more or better technical information in solving your engineering problems, Such information relates to the design and rating of heating and ventilating equipment, and to methods of installation which will secure certain specified results.

The Committee on Research is conducting a number of investigations at the present time with the sole purpose of securing fundamental technical data of use and value to the engineer, the manufacturer and the public interested in heating and ventilation. The work now in progress is summarized in the accompanying preprint of an article entitled, A NEW ERA IN A.S.H.V.E. RESEARCH BEGINS, appearing in the June Journal of the Society.

Will you, therefore, be good enough to send me your suggestions concerning research problems of particular interest in your special field? Please refer to the preprint and list your problems as under three main divisions:

1. Entirely new investigations
2. Present investigations now in progress
3. Past investigations, which you feel should be reviewed or amplified

It will, of course, be impossible to undertake at once all the problems which will be suggested, but the total of such suggestions will prove a valuable index to the Committee on Research in setting up a research program of the greatest general interest and value.

Very truly yours,

C. V. Haynes, Chairman
COMMITTEE ON RESEARCH

CVH/IMM

Present Work Commended

The response was both prompt and illuminating, and the detailed and constructive replies which are arriving daily are being carefully reviewed and are to be considered by the Committee on Research as a basis for the future Research Program of the Society. It is interesting to note also that practically all of the present research projects, now in progress at the Research Laboratory in Pittsburgh and at the various co-operating institutions, are of definite interest to those who replied to Chairman Haynes' letter.

In many cases, the writers not only commended the existing Research Program, but also made detailed reference to specific projects which were of real value to them in their work.

Both Engineering and Physiological Data Desired

The data most desired, as expressed in the letters now at hand, are both of an engineering and physiological character. Naturally, the former division predominates, but there are many requests for information in the latter division as well. In fact, the rapid developments in the field of air-conditioning of occupied spaces has forced the heating and ventilating engineer to establish and seek out new and more reliable data as to the effect of the atmospheric environment on human comfort and health. Air conditioning for materials and processes in the industrial field has been largely an engineering problem from the start, except in so far as the plant operatives were involved in the application.

Objectives Sought in Requests

Under the engineering division, three obvious motives have inspired all types of requests so far received. In every case, A. S. H. V. E. members are interested in securing information with one or more of the following objectives:

1. Performance data on heating and ventilating equipment such as testing and rating various units and materials.
2. Design data for laying out and installing a complete heating or ventilation system.
3. Economic data in the form of a survey and report on "best" or most economical type of equipment or system for heating or ventilating service.

Comparative information is much in demand, based on actual installations. To just what extent the Research Program should include projects under this third objective, involving the idea of a "survey and report," may be open to question, but there can be no question whatever about the importance of such a service. It is a matter to which the Committee on Research will give its serious attention.

Since, in many cases, the requests involve performance, design, and economic data in the same proposal, no attempt has been made to list the proposals under these three types. If this should be done there would be many repetitions, which have been avoided by classifying the proposals under about 20 general subjects and indicating the number of proposers for each item.

Under the physiological division, two obvious types of requests occur, with the following objectives in view:

1. Standards of ventilation or air-conditioning for a wide variety of service, from hospitals to libraries.

2. Essential factors affecting human comfort and health, such as air temperature, humidity, air motion, wall surface temperatures and ionization. The relative importance of these factors is much in demand, such as effect on occupants of convection heated as compared with radiant heated rooms.

Summarized Results

As it may be of interest to the members of the Society to know just what research problems are of the greatest present importance to those who answered Mr. Haynes' letter, the following condensation of the first 40 replies has been prepared in the form of a *SUMMARY* giving only such details as seem essential to make each item understandable. The proposers have been keyed so that each reference may be properly correlated with its origin if desired:

SUMMARY OF SUGGESTIONS

Proposer	Suggestions in Brief	Remarks
I.	Reducing valve capacities for heating at 2 lb gage pressure reduced from 50 to 100 lb steam	Proposed
II.	Economic study of the cooling of offices, residences, hotels, and apartments.	Proposed
III.	1. Heat absorbed by water legs of fire box heating boilers using gas, oil, and coal. Also heat released per cubic foot of furnace and per square foot of furnace wall.	Proposed
	2. Heat transfer steam to air for commercial fin tube heating surface. Also temperature gradient along fins.	Proposed
	3. Cooling of gases carrying water vapor where condensation occurs.	Proposed
IV.	Economic study of air conditioning office buildings applied to U. S. Bureau of Standards.	Proposed
V.	Heat emission from large surfaces; still and moving air conditions for	
	(a) building walls.	In progress
	(b) ovens and furnaces up to 700 F. surface temperature.	Proposed
VI.	1. Heat transmission of walls under fluctuating temperature conditions, including solar effects. Tests to be checked against actual buildings to establish practical formulae.	Proposed
	2. Determine vital factors in ventilation essential to health.	In progress
	3. Code for testing ferrous and non-ferrous radiators of concealed cabinet type.	In progress
VII.	1. Air temperature gradient from floor to roof in single story industrial building. Three types of heating systems: direct radiation, low velocity unit heaters, and high velocity floor type unit heaters.	Proposed
	2. Effect of high or low outlet air temperatures on	
	(a) location of unit heaters,	Proposed
	(b) outlet velocity in feet per minute.	Proposed
VIII.	1. Heat transmission of building materials.	In progress
	2. Solar effects on building roofs and walls.	In progress
	3. Infiltration of air through walls and building openings.	In progress
IX.	1. Code for testing and rating vacuum pumps for air and water capacities including standard conditions for testing and rating.	Proposed
	2. Introduction of free oxygen into heating systems through vacuum pumps.	Proposed
	3. Interested in all present investigations.	In progress
X.	1. Tests of heating boilers using oil burners.	In progress
	2. Performance tests of ferrous and non-ferrous radiators under actual heating service conditions.	In progress
	3. Testing and rating heaters for hot-water heating service.	Proposed
	4. Testing and rating of various enclosed types of non-ferrous hot-water radiators.	Proposed
XI.	An elaborate and detailed program of tests to determine various factors affecting the design and operation of automatic underfed stokers.	Proposed
XII.	An investigation of coal stokers for domestic heating purposes making a distinction between stokers burning anthracite and bituminous coals.	Proposed
XIII.	More data on insulating building walls including conductivities of recent insulating materials.	In progress
	2. Investigation of human comfort in rooms with the wall surfaces colder than the room air. Suggested wall surface temperatures of 40, 60, and 70, with the room air at 70 F.	Proposed
XIV.	A survey and report on methods of heating multi-story library buildings:	
	(a) with book racks independent of the floor construction.	Proposed
	(b) with the racks sealed off at each floor level.	Proposed
	This survey should also include various aspects of the best air conditions for books and occupants in such buildings.	
XV.	1. Investigation of the flow of air through window and door screens in dwellings and other buildings in connection with cooling such buildings in the summer.	Proposed
	2. A study of the effect of various air temperatures on patients suffering from hay fever, asthma, and other similar ailments, in other words, is temperature a factor independent of the dust or pollen content of the air?	Proposed
XVI.	Investigation of the proper method of insulating a double roof construction in which there is a false ceiling below the roof proper. Where is the best place to apply the insulation in order:	
	(a) to secure the maximum insulating effect,	Proposed
	(b) to avoid condensation either with or without ventilation of the air space between ceiling and the roof deck. If such a space is ventilated, how does it affect the value of the roof insulation? If such an insulated space is not ventilated, what will be the effect on condensation?	Proposed
XVII.	A study of steam pressure regulating and reducing valves,	
	(a) allowable pressure drop in such valves,	Proposed
	(b) suitable materials for use in such valves at high temperatures.	Proposed
XVIII.	1. Ventilation of elevator machine rooms; requirements of such service and best method of ventilation.	Proposed
	2. Economic study of heating buildings by electricity especially in milder climates.	Proposed
	3. Tests of oil burners in heating boilers when operating at low ratings.	In progress
	4. A study of materials or coatings which will withstand the corrosion that occurs in return lines. (See a survey by the N. D. H. A. on this subject.)	Proposed
XIX.	1. Testing and rating of concealed convection type radiators with gravity and fan circulation.	In progress

SUMMARY OF SUGGESTIONS

Proposer	Suggestions in Brief	Remarks
	2. Comparison of air temperature gradients, ceiling to floor for concealed convection type copper radiators and ordinary cast-iron radiators.	In progress
	3. A study of the electrolysis between copper tubes and cast-iron end section of copper radiators in both hot-water and steam-heating systems.	Proposed
XX.	1. Miscellaneous data covering the design of all-mechanical warm-air heating systems for residences.	Proposed
	2. Interested in four of our present investigations.	In progress
XXI.	Comparative temperature studies in the furnaces of fire-box boilers as compared to boilers with refractory walls.	Proposed
XXII.	Comparative study of the heating efficiency (steam consumption) and "heating effect" of unit heaters versus cast-iron radiation for heating industrial buildings based on field tests in actual buildings.	Proposed
XXIII.	Relative amount of heat transmitted by radiation, convection, and conduction in typical cast-iron boilers burning coal, oil and gas.	Proposed
XXIV.	1. Various data relating to the design of all-mechanical warm-air heating systems for residences with special reference to heating efficiency, power consumption, noise, and allowable air velocities.	Proposed
	2. Interested in practically all present investigations.	In progress
XXV.	1. Design and installation data for all-mechanical warm air heating systems including register locations, air temperatures, and velocities through inlet registers and most economical combination of inlet air temperature and total air volume introduced.	Proposed
	2. Interested in all present investigations.	In progress
XXVI.	1. Testing and development of devices for controlled humidity in occupied rooms during the heating season.	Proposed
	2. Interested in heat transmission of wood as used in building construction.	In progress
XXVII.	1. A comparative study of the gravity convection heaters with reference to location, steam consumption, heating effect, regulation and humidity conditions.	In progress
	2. Proper methods of ventilation and temperature control in elevator machine and control rooms.	Proposed
	3. A study of the effect of smaller pipe sizes with various steam pressures on so-called orifice heating systems.	Proposed
	4. Accurate tests on steam consumption with several types of temperature control apparatus in same building.	Proposed
XXVIII.	Tests on the insulating value of various thicknesses of woven asbestos products for use on high pressure steam line and equipment.	Proposed
XXIX.	An investigation of sealed or closed hot water systems using various pressure devices. The results of this investigation should be incorporated in a code governing the use of cast-iron boilers and radiators in such systems.	Proposed
XXX.	Miscellaneous problems relating to the installation and operation of under-ground piping systems including the conductivity of various kinds of ground. (Suggests possibility of a joint investigation with the Manufacturers' Division of the *National District Heating Association.*)	Proposed
XXXI.	Design and development data relating to all-mechanical furnace heating systems using coal, oil, or gas as fuel. Such information as desired for units completely equipped with all accessories for air conditioning under automatic control.	Proposed
XXXII.	An investigation or report on the effect of various water pressures on the operation of closed hot water storage heaters containing steam coils at constant temperature and pressure.	Proposed
XXXIII.	Investigation of the heating effect of radiators with special reference to the volume of air moved as compared with the radiating effect of the unit. This study would involve the measurement of the quantity of the air moved or circulated by the radiator.	Proposed
XXXIV.	Performance test of closed, high temperature hot water heating systems with forced circulation. Special consideration to be given to concealed radiation of the convection type and its adaptation to hot water heating. Comparative data to be desired on the operating characteristics of vapor steam systems and forced hot water systems for heating the same building.	Proposed
XXXV.	A study of the adaptation of an all-mechanical warm air heating system so as to provide for periodical gravity circulation of the system with the fan not running. (Offers the benefit of own experience and practice with this particular type of system.)	Proposed
XXXVI.	Design and installation data for all-mechanical warm air heating systems in residences. These data should include both heating and cooling service.	Proposed
XXXVII.	1. Comparative study of concealed gravity convection heaters and exposed cast-iron radiators to determine whether or not condensation tests accurately indicate the "heating effect" of these two different types.	In progress
	2. Interested in two present investigations, (a) flow characteristics of copper tubing for steam and hot water heating, (b) the performance characteristics of non-ferrous radiators with extended fin surface.	In progress

Basis for Selection of Projects

A further analysis of these replies under about 20 general subject classifications has been made and shows how many proposers have suggested the same research project. This latter information is, of course, desired by the Committee on Research, but in reaching a decision as to which of the new projects shall receive first consideration, the basic character of the requests as suitable research projects, and the probable technical interest of such research to the entire A. S. H. V. E. membership, must also receive the attention of the Committee.

Each mail brings in additional suggestions and the entire list will be classified and considered at the next meeting of the Committee on Research.

The response of Society members has been enthusiastic and indicates the members' appreciation of the value of the Society's leading activity and their eagerness to participate. This is most gratifying to the Committee on Research.

A.S.H.V.E. Members Select Candidates for F. Paul Anderson Medal

THE Committee on Award of the F. Paul Anderson Medal extends an invitation to all members of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, to submit the names of candidates for the first award of the F. Paul Anderson Medal, to be made at the 38th Annual Meeting of the Society January, 1932.

The conditions are briefly as follows:

1. Any member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in good standing during 1931 is eligible for nomination.
2. Each nomination, sent to the Committee on Award must be accompanied by a statement of the candidates accomplishments specifically mentioning the work done or services performed and indicating in what way such work or services have been outstanding in the field of heating, ventilating or air conditioning, during any year or years prior to 1931.
3. All nominations must be in the hands of the Committee on *November* 30, 1931, and are to be addressed: Committee on Award of the F. Paul Anderson Medal, c/o Secretary, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, 51 Madison Ave., New York, N. Y.

The Committee on Award of the F. Paul Anderson Medal was created by resolution of the Council June 22, 1931, under regulations adopted January 26, 1931, and consists of the three last presidents, the first and second vice-presidents and one member at large, namely:

A. C. Willard, *Chairman*
Thornton Lewis — F. B. Rowley
L. A. Harding — W. T. Jones
F. C. McIntosh

At the 36th Annual Meeting in Philadelphia, January, 1930, the Society accepted the sum of $1000 presented by Pres. Thornton Lewis as an endowment to establish the F. Paul Anderson Medal.

In addressing the Council, Mr. Lewis' letter of January 13, 1930, said:

> It has seemed to me, for some time, that our Society would be benefited if we could devise some method for stimulating the younger men to take a greater interest in our work.
>
> Also that since we have no awards for meritorious service, it would be desirable to encourage the establishment of such foundations.

With these thoughts in mind I herewith take pleasure in presenting to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS a certified check for the sum of One Thousand Dollars, subject to the following conditions:

1. That this sum, together with such other sums as may be added to it, be made a permanent endowment fund, the income from which shall be used to provide an annual medal of award (accompanied by suitable certificate), to some member of the A. S. H. V. E. in recognition of work done or services performed in the field of Heating, Ventilating or Air Conditioning;
2. That the Council of the A. S. H. V. E. shall in its discretion invest the principal and administer this fund, decide itself, or by special committee appointed by it, to whom the award shall be made, and shall decide and announce to the Society at least one year in advance, the conditions upon which the award will be made:
3. While it is herein stated that this is to be an annual award, the Council of the A. S. H. V. E. shall have the power to refuse to award the said medal in any year if in its opinion no work or service worthy of the honor has been performed in accordance with the conditions which the said Council has previously set forth. Should such conditions prevail the income from the fund shall be added to the principal;
4. That all decisions of the Council relating to this award shall be made by a majority vote of all the members of the Council;
5. Should the A. S. H. V. E. go out of existence then its last Council, duly elected, shall have the power to dispose of this endowment fund in such manner and for such purpose as may best promote the art of heating and ventilating;
6. The name of this award shall be "The F. Paul Anderson Medal."

Following the acceptance of this gift the Council on May 19, 1930, announced the appointment of Messrs. McIntosh, Cassell and Farnham as a special committee to prepare rules of award for the F. Paul Anderson Medal.

On June 24, 1930, an additional gift of $500 was received by the Council from an anonymous donor for the F. Paul Anderson Medal fund.

During the 37th Annual Meeting in Pittsburgh, January,

1931, announcement was made by F. C. McIntosh, Chairman of the Committee on Rules of Award, that regulations had been approved by the Council and that the first award of the F. Paul Anderson Medal would be made in January, 1932, to a member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, who had performed an outstanding service for the art of heating and ventilating.

The procedure for making the award is as follows:

ARTICLE III.

MAKING THE AWARD

Section 1. Any nominations made must be carefully considered by the Committee.

Section 2. The Committee may make recommendations for award whether or not any nominations have been received.

Section 3. At the final Council meeting of a calendar year, the Committee on Award is to report its recommendations for the award of the medal and these recommendations, together with any additional nominations made by Council, are to be mailed at once by the Secretary to all Council members.

Section 4. At the same time, letter ballots bearing the names of all nominees are to be distributed to all Council members, with a statement as to when ballots are to be counted, which date is to be approximately two weeks from date of mailing by the Secretary. Unless one nominee receives a majority vote, successive balloting is to be carried on similarly until a majority favorable vote is secured for one nominee. Names of nominees receiving only one vote shall be dropped from succeeding ballots. The result of each balloting is to be given to all Council members.

Section 5. Council has the power to decline to make any award for any year.

Section 6. No award may be made except in conformity with announcement made for the year.

Section 7. Presentation shall be made publicly by the President of the Society, at a time and place agreed upon by the recipient and the Officers of the Society.

ARTICLE IV.

THE F. PAUL ANDERSON MEDAL

Section 1. A gold medal to be known as the "F. Paul Anderson Medal" may be awarded annually to a member of the Society, together with a suitable certificate, in recognition of work done or services performed in the field of Heating, Ventilating or Air Conditioning. If funds are available, a bronze replica may be given with the medal.

The F. Paul Anderson Medal was designed by John Swanson and carries a likeness of Dean Anderson on one side and handsome figures typifying heating and ventilating on the reverse side. The medal is 1½ in. in diameter and is cast in 18K gold.

The Committee on Award desires to have the cooperation of all members of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in selecting candidates for this award and nominations should be in the hands of the Committee by *November* 30, 1931.

Summer Meeting of I. H. V. E.

The attendance, at the summer meeting of the *Institution of Heating and Ventilating Engineers,* Great Britain, held at the Hotel Majestic, Harrogate, June 23rd, was the largest in the history of the organization, with 166 members and guests present.

Pres. Sam Fox presided and welcomed the members to Yorkshire, after which the Secretary announced the names of the candidates elected to the various grades.

The reports of several committees were presented and then W. W. Knobs outlined the progress made by the Research Fund Committee. It was reported that the British Government had offered assistance in promoting the research work of the Institution and the President urged the members to do their duty in backing the enterprise to the best of their ability.

A. F. Dufton presented a paper on Radiant Heat, illustrated with lantern slides, and an interesting discussion followed.

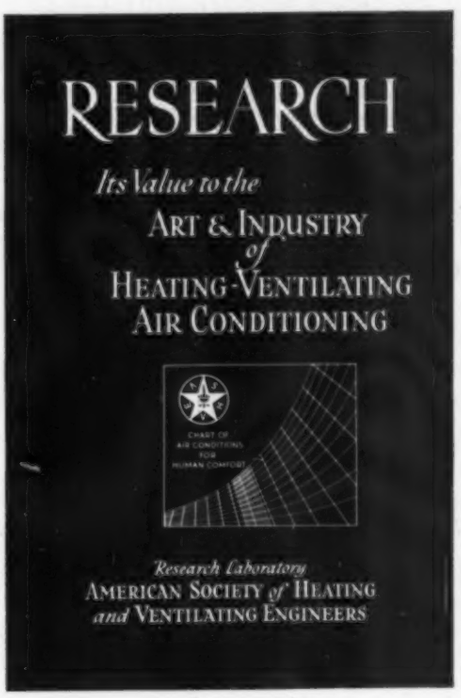
The Haden Golf Cup was jointly won by Messrs. Alden and Hoare with net scores of 76.

The members and ladies enjoyed a motor trip to Fountains Abbey, and in the evening a banquet was held with Lt. Col. E. Kitson Clark, T. D., M. A., President of the *Institution of Mechanical Engineers,* as the principal speaker.

On June 24 a group photograph was taken and this was followed by a motor drive and luncheon at the Devonshire Arms Hotel.

Research Booklet Sent to Society Members

The value of research to the art and industry of heating and ventilating is outlined in a booklet entitled, RESEARCH, prepared by the Committee on Research of the AMERICAN SOCIETY



OF HEATING AND VENTILATING ENGINEERS, and mailed to the membership August 3 with a letter from Chairman C. V. Haynes.

This striking booklet, with a black cover printed in silver, tells the purpose of the Society's Research work, explains the method of administration; tabulates the expenditures made annually since 1919; outlines the method of co-operation with other institutions; lists the 16 present activities of the Research Laboratory; gives a brief and inspiring history of the Laboratory's creation; explains the method of distributing data compiled by the Research Laboratory and includes a list of the Laboratory publications.

This list of papers is a truly impressive one and shows how varied are the subjects of investigation and how much has been contributed by the Society to the comfort and enjoyment of mankind.

These papers give an accurate indication of the Laboratory's record of progress and it is easy to judge the practical value of the accumulated data.

The booklet is profusely illustrated with photographs of work in progress both at the Research Laboratory located in the United States Bureau of Mines, Pittsburgh, and the various co-operating institutions.

A. S. H. V. E. members will be proud of the service rendered by the Research Laboratory, founded in 1919 and supported so generously since that time both by the membership of the Society and the heating and ventilating profession and industry.

Local Chapter Reports

Cleveland

The Cleveland Chapter of the A. S. H. V. E. has completed its activities for the 1930-31 season, and the officers elected to hold office during 1931 are as follows:

President—W. E. Stark.
Vice-President—C. J. Deex.
Treasurer—H. M. Nobis.
Secretary—R. G. Davis.
Board of Governors—F. H. Morris, C. Gottwald, S. H. Givelber.

The subjects of the papers discussed at the meetings during the year and the names of the speakers are:

October 1930—A. P. Kratz: Research in House Heating (Joint meeting with the *A. S. M. E.*)

December 1930—F. I. Raymond: Scientific Heat Control for Large Buildings.

February 1931—Webster Tallmadge: Zone Heating of Buildings.

March 1931—H. F. Marshall: An Illustrated Discussion of Steam Consumption of Heating Systems.

June 1931—W. E. Stark: What is Doing in Gas Today.

Kansas City

The Kansas City members completed their activities for the year with the June meeting, when they elected the following officers to serve for the year 1931-32:

President—J. M. Arthur.
Vice-President—W. A. Russell.
Secretary—D. D. Zink.
Treasurer—Emil Haas.
Board of Governors—E. K. Campbell, T. L. Dawson and J. G. Lewis.

The first meeting of the season was held in September 1930, and it was at this meeting that C. A. Burton gave an enlightening talk on his experience as a member of the Water Supply Committee in conjunction with the Kansas City 10-year plan, which was one of the most interesting talks of the evening. N. W. Downes reported on the Semi-Annual Meeting of the Society, while E. K. Campbell, Jr., gave an able discussion on the Willard and Kratz paper which was presented at the summer meeting.

The next meeting was held in December, and Prof. J. V. Martenis gave an interesting talk, illustrated with suitable slides, depicting the development of heating and ventilating. The civic feature was an address by E. H. Gill, Industrial Division of the Kansas City Chamber of Commerce, who outlined the steps taken by that body to relieve unemployment.

The January meeting was made quite interesting by a moving picture, furnished by the Aluminum Co. of America, which gave a description of the construction of a great dam across the Saguenay River at Arvida, Quebec. Another talk was given by E. E. Howard, and was devoted to the growing importance of engineering in all phases of life, and the increasing obligation of the engineer to society as a whole as a result of it. He also emphasized the ethical side of the engineering profession.

When the members met for the February meeting, N. W. Downes reported on his activity in behalf of the Chapter, regarding the proposed legislation designed to raise the standard of engineers and engineering by a registration law for professional engineers. This subject brought a lively discussion, and suitable resolutions were drawn up to carry to the Legislative body the feelings of the Society in regard to this proposed legislation. Following Mr. Downes' talk Prof. A. H. Sluss of the University of Kansas invited the Chapter to be his guests at the University to hear Dr. S. P. Grace, Bell Telephone Laboratories, New York City, who demonstrated the latest developments in telepathy.

One of the most interesting meetings of the year was held at the University of Kansas in April, when the members were addressed by Pres. W. H. Carrier, who spoke on Servicing the Human Power Plant, which was thoroughly enjoyed. President Carrier also touched on the broader aspects of the A. S. H. V. E. in regard to its research work. The members and guests were not only well entertained, but found President Carrier's talk very instructive.

The final meeting of the Kansas City Chapter, held in June, was combined with business and social functions. Following the swimming and boating, which preceded an excellent dinner, Professor Cady, Kansas University, delighted everyone with his illustrative lecture on Liquid Air.

Before adjourning to the ballroom, where a nine-piece orchestra was in readiness, the annual election of officers took place.

Illinois

From October 1930 to May 1931 a series of monthly meetings were held by the Illinois Chapter of the Society, and were conducted under the leadership of the following officers, who held office from October 1930 to October 1931:

President—A. G. Sutcliffe.
Vice-President—J. J. Aeberly.
Secretary—C. W. DeLand.
Treasurer—C. W. Johnson.
Board of Governors—H. P. Reid, R. B. Hayward, H. G. Thomas.

Secretary DeLand has reported an average attendance of 100, and gives a complete list of the meetings as follows:

October 1930—Pres. L. A. Harding: Utilization of the Sun's Energy. F. C. Houghten: Effects of Solar Radiation on Heating and Air Conditioning Problems.

November 1930—Joseph Harrington: Stokers, and Their Part in Smoke Abatement and Fuel Economy.

December 1930—J. M. Robb: Centralization of Responsibility in Heating Practice.

January 1931—Dr. E. V. Hill: Aerology and Eternity; the Physiological Possibility of Air Conditioning.

February 1931—H. G. Thomas: Delivered President Harding's paper on Heat and Its Effect on Our Cultural Relationship.

March 1931—Prof. A. C. Willard: Ventilation Investigation for the New Chicago Subway.

April 1931—Pres. W. H. Carrier: Servicing the Human Power Plant.

At the April meeting the Illinois Chapter had as its guests the Officers of the Society, members of the Council, the members of the Committee on Research, and on Ventilation Standards, and six past-presidents of the Society. There were over 200 members and guests at this particular meeting.

The May meeting was in the form of a golf tournament, which, from all reports, proved very successful.

The annual meeting of the Chapter is scheduled for October 12, 1931, at which time the election of officers for 1931-32 will take place.

Michigan

With an attendance averaging about 80 the Michigan Chapter held eight meetings during the year, beginning with the October 1930 and ending with the May 1931, at which time the following officers were elected to serve the Chapter for the year 1931-32:

President—George Giguere.
Vice-President—W. J. Whelan.
Treasurer—E. H. Clark.
Secretary—G. D. Winans.

Board of Governors—N. B. Hubbard, L. L. McConachie, H. E. Paetz.

The topics discussed at these various meetings are:

October 1930—Count Alfred Von Nienschowski: Experiences During the World War. G. R. Thompson: Building Activities of State of Michigan.

November 1930—Pres. L. A. Harding: Utilization of the Sun's Energy. F. C. Houghten: Solar Energy and Its Effect on Heating and Ventilating Problems. A. V. Hutchinson: Account of visit to Chapters on West Coast.

December 1930—E. R. Shaw: U. S. Chamber of Commerce.

January 1931—R. S. M. Wilde: Description of Mechanical Equipment of Cranbrook School for Boys. P. S. Russel: Oil and the Requirements Necessary for Its Proper Combustion. A. C. Hann: Developments in Practice and Devices Because of Use of Oil.

February 1931—S. S. Sanford: Modern Methods of Saving Heat in Buildings. J. H. Walker: Modeling as Applied to Pipe Design and Piping for Steam at 1,000 Deg.

March 1931—Dr. Lent Upson: Shall We Abolish Government?

April, 1931—Prof. A. C. Willard: Recent Research in Heating and Ventilation at the University of Illinois. E. B. Langenberg: Necessity of Taking Advantage of Results of Research.

The May 1931 meeting was devoted to golf and the election of officers, followed by dinner. At this meeting the Michigan Chapter was honored by having among its guests, Dean F. Paul Anderson, Lexington, Ky., past-president of the Society.

Minnesota

Many interesting meetings were held during the year by the members of the Minnesota Chapter, as reported by A. B. Algren, secretary of the Chapter.

A brief outline of the subjects discussed, together with the names of the speakers, follows:

October, 1930—Pres. L. A. Harding: Utilization of the Sun's Energy. F. C. Houghten: Effect of Solar Radiation on Heating and Air Conditioning Problems. C. W. Farrar: Membership and Society Attendance.

November, 1930—Dinner-Dance, which was well attended and proved a most successful and enjoyable party.

December, 1930—H. R. Harris: Electric Motors and Controls as Applied to Use of Heating and Ventilating Industry. D. M. Forfar: Membership—Commenting on splendid work done by A. J. Huch and Albert Buenger, membership captains.

January, 1931—C. E. Lewis: Humidity Control and Human Comfort. Carbon Monoxide, a U. S. Bureau of Mines motion picture film was presented.

February, 1931—Special Dinner Meeting. M. S. Wunderlich: Reported on officers of Society elected at Annual Meeting. Prof. F. B. Rowley: Work being carried on at Research Laboratory of A. S. H. V. E. Albert Buenger and E. F. Jones: Outlined the many unique heating and ventilating problems in connection with the design of systems for the Northwest airways and shops and 109 Aero Squadron. An inspection trip was made of the two installations at the St. Paul Municipal Airport.

March, 1931—John F. Hale: Progress made in Heating and Ventilating Industry since the early nineties.

April, 1931—Pres. W. H. Carrier: Servicing the Human Power Plant.

May, 1931—Afternoon was devoted to golf, followed by dinner and the election of the following officers:

President—H. E. Gerrish.
Vice-President—W. F. Uhl.
Secretary-Treasurer—A. B. Algren.
Board of Governors—D. M. Forfar, M. H. Bjerken.

One of the most interesting meetings ever held by the Minnesota members was that held in April, when Pres. W. H. Carrier and A. V. Hutchinson, secretary of the Society, were the honored guests.

New York

The New York Chapter of the A. S. H. V. E. held monthly meetings from October to May during the 1930-31 season. During this time the chapter had the pleasure of hearing technical addresses by L. A. Harding and W. H. Carrier during their terms as president.

One inspection meeting was held, and the final meeting of the season was the biggest dinner and entertainment gathering ever held by the New York Chapter. The complete list of meetings is as follows:

October 20, 1930. James H. Scarr—Weather Bureau.

November 17, 1930. L. A. Harding, President, Utilization of the Sun's Energy. F. C. Houghten, Effect of Solar Radiation on Heating and Air Conditioning Problems.

December 15, 1930. Heating and Ventilating of the Windowless "City of Light," at the Westinghouse Lighting Institute. Talks by the staff.

January 19, 1931. Carbon Monoxide—The Unseen Danger. A Department of Commerce film.

February 16, 1931. Samuel L. Goodwin, Recent Trends in Air Conditioning for Theatres.

March 16, 1931. Col. Elliott H. Whitlock, Progress in Smoke Abatement Work.

April 20, 1931. Pres. Willis H. Carrier, Servicing the Human Power Plant.

May 18, 1931. Dinner and Entertainment with members and friends who made this a very large party.

The average attendance at the technical meetings during the season was 79, according to the report of Secretary W. A. Swain.

The officers now serving the Chapter are as follows:

President—Russell Donnelly.
Vice-President—Louis T. M. Ralston.
Treasurer—F. E. W. Beebe.
Secretary—W. A. Swain.
Board of Governors—Alfred J. Offner, Albert L. Baum, Raymond Newcomb.

Western New York

The Western New York Chapter's activities during the past year were very successful both from a standpoint of attendance and character of papers presented.

Among the notable achievements of the past season was the presentation by the A. S. H. V. E. of a watch to Pres. Joseph Davis, for the largest number of new members secured from October to January.

The regular meetings of the Chapter were held at the Hotel Buffalo on the second Monday of the month. Each meeting consisted of a dinner, at which the speaker was the guest, some entertainment, a brief business session and the presentation of the paper, which was followed by an open discussion on the subject.

A summary of the meetings during the year is as follows:

October, 1930—Miss Margaret Ingels: Doing Something About the Weather.

November, 1930—Dr. James Aston: Development of the Aston Process of Wrought Iron Manufacture.

December, 1930—D. E. French: Standard Code for Testing and Rating Steam Unit Heaters.

January, 1931—J. A. Curtin: Marvels of the Sky, particularly the Moon.

February, 1931—Annual Dinner-Dance and Installation of Officers.

March, 1931—Annual Smoker at the Buffalo University Club.

April, 1931—C. J. Lundvall: Manufacture of Tubing from Ancient Greek and Roman Era up to Present Day Methods.

May, 1931—Annual Kentucky Night with Dean Anderson and his seniors as the Chapter guests. Pres. L. A. Harding: Synopsis of Utilization of Sun's Energy.

June, 1931—Dinner-Dance at Automobile Club, which closed the Chapter activities for the season.

The officers elected to serve the Western New York Chapter during the 1931-32 season are as follows:

President—Joseph Davis.

First Vice-President—M. C. Beman.

Second Vice-President—D. J. Mahoney.

Treasurer—W. F. Johnson.

Secretary—J. J. Landers.

Board of Governors—R. T. Coe, L. A. Harding, C. W. Farrar, Hugo Hutzel, W. G. Fraser, Ros Farnham, M. S. Jackson, C. A. Evans, O. K. Dyer, F. H. Burke.

Program Committee—F. H. Burke, *Chairman;* H. S. Schall, J. S. Snyder, G. A. Stephenson and R. T. Thornton.

Entertainment Committee—D. J. Mahoney, *Chairman;* H. B. Roarke and W. E. Voisnet.

Membership Committee—M. C. Beman, *Chairman;* L. N. Castin and R. P. Cook.

The enthusiastic spirit displayed by the Chapter members during the past year was most gratifying to the Officers, as indicated by Secretary Landers.

Philadelphia

A variety of interesting subjects were discussed at the meetings held during the year by the Philadelphia Chapter of the Society, and may be summarized as follows:

October, 1930—F. D. Mensing: Problems of Heating and Ventilating Work.

November, 1930—Pres. L. A. Harding: Utilization of the Sun's Energy. F. C. Houghten: Heat Absorption from Solar Radiation.

December, 1930—Dinner and meeting at the Bayuk Cigar Co., followed by an inspection of the air conditioning system installed in the plant.

January, 1931—Annual meeting of Chapter with J. D. Cassell as guest of honor and Thornton Lewis as toastmaster. Annual election of officers.

February, 1931—Lee P. Hynes: Heating by Electricity.

March, 1931—R. V. Frost: The Combustibility of Coal.

April, 1931—E. L. Gayhart: Requirements for Heating and Ventilating of Navy Ships. W. T. Breckenridge: Methods and Problems in Connection with Ship Heating and Ventilating.

May, 1931—Annual Chapter meeting at Tavistock Country Club, N. J. Also inspection of Warren-Webster & Co., Camden, N. J. H. F. Marshall: Steam Consumption of Heating Systems and Improved System of Steam Heating.

The officers elected for the year 1931-32 are as follows:

President—E. N. Sanbern.

Vice-President—L. C. Davidson.

Treasurer—M. F. Blankin.

Secretary—W. P. Culbert.

Board of Governors—H. G. Black, A. J. Nesbitt, J. H. Hucker.

Meetings Committee—J. H. Hucker, *Chairman;* A. A. Miller, W. R. Eichberg, C. S. Wilmot, W. F. Smith and Morris Sheffler.

Membership Committee—A. J. Nesbitt, *Chairman;* W. Ewald, H. J. Rettew, A. H. Woolston and R. E. Jones.

Committee on Publicity—C. A. Anderson, *Chairman;* B. W. Wilson.

Committee on Attendance—H. M. Patrick, *Chairman;* R. F. Hunder and A. R. Familetti.

Year Book Committee—H. Erickson, *Chairman;* H. J. Walther.

Finance Committee—R. C. Bolsinger, *Chairman;* M. F. Blankin and A. E. Kriebel.

Legislative Committee—F. D. Mensing (*1 year*), J. D. Cassell (*2 years*), and R. C. Bolsinger (*3 years*).

Nominating Committee—H. G. Black, F. D. Mensing, J. D. Cassell, R. C. Bolsinger and A. McClintock, Jr.

As indicated by Secretary Culbert, the total membership as of May, 1931, was 183, including Members, Associates and Juniors.

At the October 1931 meeting a motion picture will be shown by a representative of the Wheeling Steel Co., on the subject of Production of Sheet Metal.

The November 1931 meeting will be devoted to a general discussion between two architects, two engineers and two contractors, whose names will be announced later.

Ontario

The Ontario Chapter of the A. S. H. V. E. completed its season with four meetings held during the year from October, 1930 to April, 1931.

The officers elected to serve during this period were:

President—H. J. Church.

Vice-President—H. S. Moore.

Secretary-Treasurer—H. R. Roth.

Board of Governors—J. S. Wood, W. H. Boddington, W. R. Blackhall, M. Barry Watson.

The first of these meetings was held in October, and F. C. Houghten, Director of the Society's Research Laboratory, Pittsburgh, was the principal speaker of the evening. He delivered an address on Recent Developments in the Research Laboratory and Their Applications.

The next regular meeting was held in January, and consisted of an inspection of the new Bank of Commerce Building in Toronto. M. F. Thomas, consulting engineer for this modern skyscraper, arranged for this inspection and after dinner gave a short talk on the mechanical equipment of the building.

In March the next meeting of the Chapter was held, and it was at this time Esten Bolling, New York City, spoke to the members and guests on An Engineered Home Makes Home Sweet Home. This address dealt with an air conditioning experiment in Mr. Bolling's home.

The last of the meetings was held in April, with James Govan as the guest speaker. Mr. Govan selected as his topic, How Soon Will a Reduction of 60 per cent in the Heating Plant Equipment be Possible? His talk was devoted chiefly to the various experiments in the insulating of the walls in homes and industrial buildings.

The officers elected for the year 1931-32 are as follows:

President—H. S. Moore.

Vice-President—J. S. Wood.

Secretary-Treasurer—H. R. Roth.

Board of Governors—H. J. Church, J. S. Paterson, W. H. Boddington, W. M. Shears.

Attendance Committee—J. S. Wood, *Chairman.*

Membership Committee—G. Cole, *Chairman.*

Entertainment Committee—E. Gauley, *Chairman.*

Pittsburgh

The Pittsburgh Chapter of the A. S. H. V. E. completed a successful season, with monthly meetings beginning with the October 1930 meeting and ending with the May, 1931.

October, 1930—Percy Nicholls: Some Notes on the Burning Characteristics of Fuels in Heating Furnaces.

November, 1930—Pres. L. A. Harding: Utilization of the Sun's Energy. A. V. Hutchinson: Plans for Thirty-Seventh Annual Meeting.

December, 1930—Prof. F. B. Rowley: Research and Insulation; Election of Officers.

January, 1931—Discussion on plans for Annual Meeting of the Society, followed by miniature golf and dancing.

March, 1931—T. G. Estep: Small Stoker for Domestic Heating Plants.

April, 1931—Miss Margaret Ingels: Principles and Application of Air Conditioning for Residences and Public Buildings.

May, 1931—S. E. Dibble: Concealed Radiation: Types, Construction, Capacities.

Each meeting was preceded by dinner, with a comparatively large attendance. As indicated by Secretary Blackshaw, the discussions which invariably followed the addresses during the meetings have proved very profitable to the members and guests.

The officers elected at the December meeting for 1931 are:

President—F. C. Houghten.
Vice-President—R. B. Stanger.
Treasurer—H. B. Orr.
Secretary—J. L. Blackshaw.
Board of Governors—W. W. Stevenson, H. Lee Moore, C. W. Wheeler.

President Houghten appointed the following chairmen of the standing committees:

Program Committee—F. A. Gunther.
Publicity Committee—F. C. McIntosh.
Membership Committee—S. H. Harper.
Attendance Committee—G. M. Comstock.

St. Louis

The 1930-31 season of the St. Louis Chapter opened with the September 1930 meeting, and continued its monthly meetings through June, 1931.

The topics discussed and the speakers at these meetings were as follows:

September, 1930—Annual golf tournament and dinner-dance.

October, 1930—Pres. L. A. Harding: Utilization of the Sun's Energy. F. C. Houghten: Solar Radiaton.

November, 1930—C. G. Buder: Automatic Firing. E. E. Carlson: Self Contained Temperature Regulation. E. B. Evleth: Electrical Temperature Regulation. C. R. Davis: Pneumatic Temperature Control.

December, 1930—F. H. Gaylord: Heat Thief, and Controlled Heat (motion pictures).

January, 1931—G. W. Hopkins: Certified Heat.

February, 1931—R. J. Smith: Concealed Radiation.

March, 1931—C. L. Bailey: Films pertaining to the making of alloy steel.

April, 1931—E. A. Jones: Application of Gas Heating.

May, 1931—Rodney Smith: Evolution of School Heating and Ventilating.

June, 1931—G. C. Wassal: Air Transportation (with a trip over the city in a Ford tri-motor plane).

The 1931 officers of the St. Louis Chapter are as follows:

President—R. M. Rosebrough.
First Vice-President—J. M. Foster.
Second Vice-President—C. A. Pickett.
Treasurer—A. L. Walters.
Secretary—C. R. Davis.
Board of Governors—E. A. White, F. J. McMorran, George Myers, L. Walter Moon.

Southern California

A successful season was brought to a close by the Southern California Chapter of the A. S. H. V. E., with the following men as officers and members of the committees:

President—O. W. Ott.
Vice-President—F. R. Winch.
Secretary—H. B. Keeling.
Treasurer—Leo Hungerford.
Board of Governors—J. B. Armitage, H. A. Nelson, L. H. Polderman.
Program Committee—J. M. LaMontagne, *chairman;* A. R. Johnson, Robert Kratz, G. E. Clancy and H. H. Simonds.
Membership Committee—A. H. Berg, *chairman;* Harry Bremser, R. M. Gunzel and H. L. Warren.
Attendance Committee—F. H. Hilliard, *chairman;* Lee Updegraff, Fred Esser and George Horton.
Meetings Committee—H. A. Nelson, *chairman;* W. E. Schourup and C. M. Barker.
Publicity Committee—L. H. Polderman, *chairman;* H. B. Lauer and H. B. Keeling.
By-Laws Committee—C. S. Anderson, *chairman;* J. B. Armitage and J. F. Park.

At the first meeting of the year there were 54 names on the Chapter roll and at the last gathering there were 64. During that time four members moved out of the state, one died, one resigned from the Society, one resigned from the Chapter and one was dropped. The net gain in membership was 18.

The meetings of the Chapter were dinner meetings held the first Tuesday after the tenth of the month at the Masonic Club.

The following list gives the speaker and subject at each meeting:

October 1930—A. V. Hutchinson: Society Affairs.

November 1930—H. P. Etter: Pipe Welding.

December 1930—David Hall: Evolution of Cooling Systems Pertaining to Electric Machinery.

January 1931—Leo Hungerford: Heating and Ventilating Theatres from the Owner's Viewpoint.

February 1931—F. C. Houghten: The Research Laboratory.

March 1931—Pres. W. H. Carrier: Servicing the Human Power Plant.

April 1931—Harrold Doolittle: The Edison Building.

May 1931—Russell McCullough: Sound Pictures.

June 1931—Dr. Vern O. Knudsen: Acoustics in Air Conditioning.

In addition to these there were two field meetings, one in February, when the Gas Exhibit of Los Angeles served a buffet luncheon at its building for the Chapter members and guests, and the other was in March, when the Carrier Engineering Corporation's Los Angeles office conducted a tour through a theatre installation and reserved seats for the performance.

Wisconsin

The Wisconsin Chapter of the Society held their 1930-31 meetings under the supervision of the following officers:

President—J G. Shodron.
First Vice-President—E. A. Jones.
Second Vice-President—V. A. Berghoefer.
Secretary—H. F. Haupt.
Treasurer—Martin Erickson.
Board of Governors—G. L. Larson, W. F. Noll, F. G. Weimer.

The Chapter members met at dinner on the third Monday of every month at the Ogden Cafe, with an approximate attendance of 20 members, out of a total of 40. The meetings were as follows:

September 1930—Nominating Committee announced Election of Officers.

October 1930—Pres. L. A. Harding: Utilization of the Sun's Energy. F. C. Houghten: Accomplishments at the Research Laboratory. C. W. Farrar: Membership.

November 1930—S. L. Wheeler: What We Should Know About Patents.

December 1930—Prof. F. B. Rowley: Insulation.

January 1931—E. A. Jones: Development of Gas Heating; also Tests and Requirements of the A. G. A. Laboratory.

Due to a change in meeting night there was no meeting in February. The next meeting was held on March 2nd, and it was at this time that the proposal of the Chapter inviting the Society to Milwaukee for the 1932 Semi-Annual Meeting was brought up for serious and enthusiastic discussion.

A second meeting was called on March 23rd, at which a very interesting presentation of papers read at the Annual Meeting of the Society in Pittsburgh was made by able local Chapter members as follows:

Ernest Szekely: Off-Peak System of Electric Heating, by Elliott Harrington. Smoke and Dust Abatement, by M. D. Engle. Ventilation of the Holland Tunnel, by A. C. Davis.

E. A. Jones: Heating Effect of Radiators, by R. V. Frost. Air Pollution from the Engineers' Standpoint, by H. B. Meller.

J. G. Shodron: Welded Piping for Building Heating Systems, by F. G. Outcalt.

April 1931—Devoted entirely to a general discussion of the report of the special 1932 Summer Meeting Committee.

May 1931—Discussion on 1932 Summer Meeting continued: plans for annual Chapter picnic completed.

June 1931—Annual picnic at Wulffs Island, which closed the activities of the Chapter for the season.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 11 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some members by September 10, 1931, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
BUSCH, PHILIP J., Salesman, American Radiator Co., Detroit, Mich.	W. W. Bradfield William Kersjes	W. J. Carroll F. C. Purcell
CHASE, CHAUNCEY L., Gibbs & Hill, New York, N. Y.	H. L. Alt W. S. Gaylor	T. W. Reynolds A. J. Offner
DYKMAN, ORRIE JOHN, Heating & Plumbing Contractor, 818-24 Lake Michigan Dr., Grand Rapids, Mich.	W. W. Bradfield W. J. Carroll	H. E. Paetz A. P. Darlington
FINCH, STANLEY B., Supervisor, Architects and Builders Division, Brooklyn Union Gas Co., Brooklyn, N. Y.	*W. E. Bolte *G. C. Beck	*H. Behrman *E. J. Devlin (**A.S.M.E.* members)
LAMMERS, CORNELIUS, Superintendent, Glendon A. Richards Co., Grand Rapids, Mich.	W. W. Bradfield W. J. Carroll	William Kersjes F. C. Purcell
MARSHALL, ORVILLE D., Sales Engr., Illinois Engrg. Co., Chicago, Ill.	J. C. Matchett W. W. Bradfield	F. C. Purcell W. J. Carroll
MORTON, CHARLES H., 524 Murray Bldg., Grand Rapids, Mich.	W. W. Bradfield William Kersjes	F. C. Purcell W. J. Carroll
OURUSOFF, LEON, Engr., Washington Gas Light Co., Washington, D. C.	C. H. Flink H. L. Whitelaw	G. M. Maier E. R. Downe
TAZE, DONOVAN LONG, Dist. Mgr., American Blower Corp., Grand Rapids, Mich.	H. E. Paetz A. P. Darlington	D. J. King E. J. Parrock
TROSKE, JOSEPH JOHN, Vice-Pres. & Genl. Mgr., Vander Waals-Troske Co., Inc., Grand Rapids, Mich.	W. W. Bradfield W. J. Carroll	William Kersjes F. C. Purcell
WEINER, RAYMOND LOUIS, Gibbs & Hill, New York, N. Y.	H. L. Alt T. W. Reynolds	A. J. Offner Perry West

Candidates Elected

In past issues of the Journal of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Membership Committee and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

MEMBERS

CRANSTON, WILLIAM EMER, JR., Vice-President, Hoffman Specialty Co. of California, Pasadena, Calif.

CASEY, HUNTLEY FANT, C. M. Guest & Sons, Anderson, S. C.

HITCHCOCK, PAUL CAMERON, Pillsbury Engrg. Co., Minneapolis, Minn.

KINNEY, WILLIAM HAMILTON, Asst. Engr. of Subway Design, Bureau of Subways, Board of Local Improvements, Chicago, Illinois.

LEWIS, WILL, Munroe Furnace Supply Co., Omaha, Nebr.

MARSH, THOMAS A., Modern Coal Burner Co., Chicago, Ill.

NAYLOR, CHARLES LUZIER, Supt., Heating, Light and Power Dept., Atlantic Refining Co., Philadelphia, Penna.

VINCENT, PAUL JOSEPH, Chatard & Norris, Baltimore, Md.

ASSOCIATES

MATHIS, GEORGE A., Sales Manager, New York Blower Co., Cleveland, Ohio.

MESSENGER, JOHN R., Manager, Ventilating Dept., Universal Sheet Metal Co., Los Angeles, Calif.

SHAWLIN, WALTER C., Pres. & Genl. Mgr., Northwestern Fan & Blower Co., Milwaukee, Wis.

THOMAS, EARL E., General Electric Co., Los Angeles, Calif.

JUNIOR

HJERPE, CLARENCE A., JR., 93 Winthrop St., New Britain, Conn.

Heating · Piping and Air Conditioning EDITORIALS

Air Conditioning Prevents Accidents

It has been estimated by the Metropolitan Life Insurance Company that the indirect cost of accidents in industry is four times as great as the direct cost of workmen's compensation insurance premiums. This cost includes delayed production, at times causing loss of orders; decrease in efficiency resulting when experienced men who are injured are replaced by less experienced employes; cost of hiring and instructing new employes; spoilage and destruction of raw or finished material and damage to tools and equipment.

Accidents frequently occur from lack of proper ventilation or air conditioning. We are familiar with accidents in garages due to inhalation of carbon monoxide gas. Superintendents of large manufacturing plants are aware of the dangers that lurk in some parts of the plant from escaping gases. These are the more obvious causes of accidents.

Dye-house ventilation, which clears the atmosphere of the steam generated from the tubs, is an illustration of accident prevention by ventilation. Anyone who has been in an unventilated dye-house, stumbled into pits and tripped over trucks, readily appreciates its use.

Or take a ventilation system installed in an electroplating establishment, and designed to remove fumes generated in the vats. These fumes are poisonous, and prior to the installation of the system it was necessary to wear gas masks. It could be argued that to do away with the gas masks is a reduction in hazards, as the wearing of them certainly does not permit free vision and full cognizance of what is going on within the immediate vicinity of the wearer.

But there are many accidents and sicknesses which may reasonably be ascribed to, and in many instances are definitely proved to be, the result of bad air. Accidents are avoided in many cases by mental alertness and while the mental effects of poor air conditions are not measurable, we know from the physiological effects that they must be great.

A clearer knowledge of the underlying causes of accidents would be desirable. It is not sufficient to locate and measure hazards; more should be known about contributing conditions such as poor air and poor lighting. There is no doubt that the conditions surrounding employes have a lot to do with accidents; more specific data are needed.

Increased Net— Decreased Gross

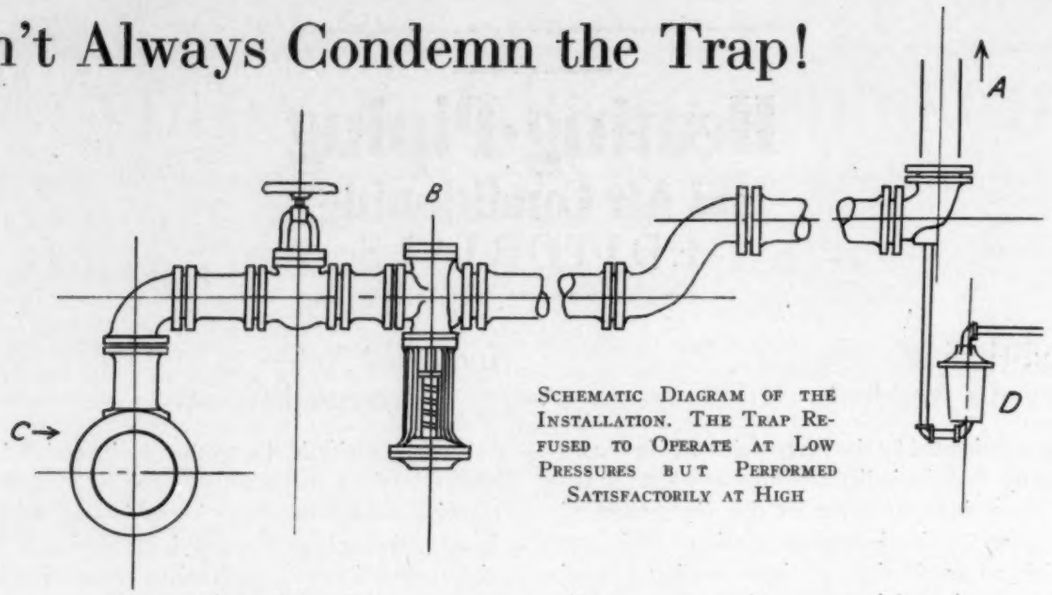
That necessity is the mother of invention is demonstrated by the managements of several concerns that are showing smaller decreases in net income than in gross income, through more careful application of the principles of selling and production. According to figures recently published in *Printer's Ink*, many companies reported decreases in net income that show smaller net declines than did gross income. Other concerns showed increases in net income, though the gross declined.

Among the former is a large corporation with a 25.4 per cent decrease in net for the first six months with a 28.5 per cent decrease in sales. An automobile concern showed an increase of 33.3 per cent in net income for the second quarter, notwithstanding a 15.6 per cent decrease in sales. Another manufacturing concern, with sales off 3.2 per cent for the first six months, enjoyed a 35.2 per cent increase in net income.

Careful economies will enable other plants to duplicate these experiences. The engineering department should not overlook opportunities for improvements in piping systems. Readers of this publication will recall how the Port Huron Sulphite & Paper Co. reduced the loss of good stock from eight to five per cent and improved quality by means of a revised white water piping system. In plants using process steam and desiring to increase the ratio of by-product to purchased power, it may be found that the answer is increased steam pipe sizes; frequently, the resulting economies have paid for the piping changes within a year. In refrigerating plants improved piping will frequently cut power costs, increase compressor capacity, and reduce the cost per ton of refrigeration. There are numerous other similar instances.

In almost every plant and building, the piping represents an important transportation system. Like every transportation system, piping must be maintained conscientiously at all times in order that production costs may be kept at a minimum.

Don't Always Condemn the Trap!



SCHEMATIC DIAGRAM OF THE INSTALLATION. THE TRAP REFUSED TO OPERATE AT LOW PRESSURES BUT PERFORMED SATISFACTORILY AT HIGH

By T. H. Rea*

A STEAM trap that works in a satisfactory manner at one pressure and refuses to open when the pressure is raised does not cause special comment, as a pressure rise of consequence will prevent the operation of most traps. The reason is that the force available for pulling the valve away from the seat is definitely limited by the weight of the float in the trap multiplied by the leverage.

For example, let F equal force available in pounds for opening valve, P equal the steam pressure, and A the area of the valve orifice in square inches. Then A must not be greater than F/P or the trap will not open. For instance, with a bucket weighing one pound and a leverage of 5 to 1, F would equal five pounds. At ten pounds pressure the largest orifice that could be used would be 5/10 or ½ sq. in. in area. If the pressure were raised to 11 pounds, the force available would not be equal to the area multiplied by the pressure and the trap would not open.

Diagram Shows Trap that Refused to Work

However, when a trap refuses to work at a low pressure but starts to operate when the pressure is raised, a lot of head scratching is done. Such a result is contrary to ordinary experience; the figure shows an installation where the traps would work at high but not at low pressure.

Steam passing through line A leading to the upper floor of a 40-story building is supplied from a district heating company's main C through reducing valve B. The initial steam pressure is 125 lb. and the reduced pressure varies from 30 to 50 lb. When there is a considerable demand for steam, the pressure will reach the lower limit and when the load is light, the pressure will rise.

At pressures ranging from 30 to 37 lb., trap D will remain tightly closed. As soon as the pressure passes the 37-lb. mark, the trap discharges condensate regularly. At first, it was assumed that the trap was defective but examination showed it to be in A1 mechanical condition. Since mechanically operated traps will work at any pressure below the maximum, it is evident that their failure to open at low pressure must be because no condensate can get to the trap.

High Steam Velocity Thought to Be Cause

One theory advanced to account for this was that the high velocity of steam—at times of low pressure and maximum demand—carries the condensate past the drips leading to the traps. While the failure of condensate to reach the trap might be due to high velocity, it seems more likely that there is no condensate to be handled.

Steam Superheated—No Condensate Formed

When steam is passed through a reducing valve, no heat is lost. Yet the temperature of saturated steam at 30 lb. is lower than the temperature of saturated steam at 125 lb. Since no heat is lost on expanding the steam, it follows that the expanded steam becomes superheated. When 100 per cent quality steam at 125 lb. pressure is expanded to 50 lb. the steam has 26.7 degrees of superheat and 41 degrees of superheat if it is expanded to 30 lb. If the 125-lb. steam is not 100 per cent quality and contains moisture, this moisture must be evaporated before the reduced pressure steam will take on any superheat. Steam of 98.5 per cent quality expanded from 125 to 30 lb. will have approximately 15 degrees of superheat but when expanded to 50 lb., there will be almost no superheat.

Even if there were some superheat in the steam right at the reducing valve, this would be dissipated by radiation losses from the steam header. Referring again to the installation shown in the diagram, it is reasonable to assume that at pressures below 37 lb., the steam is superheated up to the trap and hence there is no condensate for the traps to handle. At pressures above 37 lb., the superheat disappears, condensate is formed and the traps go into operation.

Don't Condemn Trap Always

Either superheat or high steam velocity might be responsible for the failure of condensate to reach the traps. In either case, it demonstrates that traps should not be condemned, even though circumstantial evidence would indicate that they are at fault.

*Armstrong Machine Works, Three Rivers, Mich.

Practical Piping Problems

Reclaiming Arrangement for Waste Oil

W. H. Wilson's Monthly Discussion of the Problems He Meets From Day-to-Day

Industrial plants having long oil transmission pipe lines to the various buildings and shops are usually bothered with the problem of oil leaking and finding its way to tunnels, pits and basements. This wasted oil, when allowed to pass directly into sewers or streams, is not only a money loss but is the cause of complaints from municipal authorities and game and fish wardens.

A suitable arrangement for collecting and reclaiming the waste oil, when the drainage is handled by pumps or ejectors or when it can be arranged to flow into a receiving tank, is shown by the illustration on the next page.

The tank, *A*, may be constructed of steel or concrete. A baffle partition, *B*, forms a place to receive the discharge from the sewage ejector, *C*. The baffle partition causes the sediment to remain in this compartment, where it can be conveniently removed. The partition being lower than the top of the tank, the oil and excess water flows over the baffle into the main part of the tank and the water passes out through the water overflow, *D*.

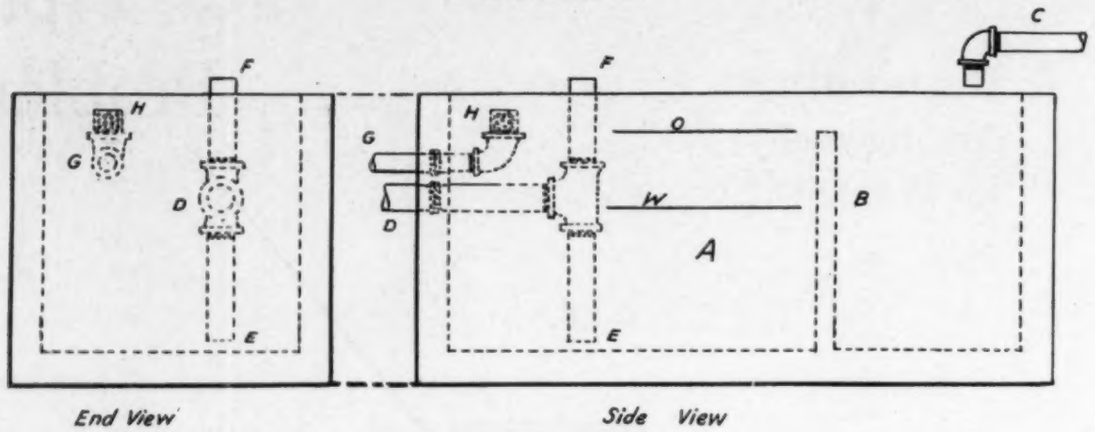
The inlet end of this overflow is submerged to within a few inches of the bottom of the tank, as shown at *E*. The overflow pipe is open at the upper end for a vent, as at *F*. This overflow should lead to a manhole or catch basin and be provided with an approved drainage connection to the sewer or waste water drainage system.

The waste oil flowing the length of the tank will be separated and remain on the surface of the water. As the oil accumulates it will pass out through the oil outlet pipe, *G*, where it can be collected, or if convenient, passed into an oil storage tank.

When the tank for such oil reclaiming equipment is constructed of concrete, care should be taken in making a tight joint where the pipes pass through the concrete end wall. Before the form is poured, a few heavy strands of hemp or flax packing wound around the pipe in the center of the form will aid materially in making an oil-and water-tight joint.

The oil outlet pipe should be provided with a suitable

THE DRAWING SHOWS A TANK AND PIPING ARRANGEMENT FOR RECLAIMING WASTE OIL FROM DRAINAGE LINES, NOT ONLY TO SAVE THE OIL, BUT TO PREVENT IT FROM GETTING INTO LAKES AND STREAMS. GAME AND FISH WARDENS REGISTER STRONG COMPLAINTS AGAINST THE PRACTICE OF ALLOWING WASTE OIL TO GET IN STREAMS. THIS ARRANGEMENT IS ALSO HELPFUL IN CASE AN OIL LINE BREAKS, OR LEAKS BADLY. THE OIL IS NOT ONLY SAVED, BUT THE LEAK IS USUALLY DISCOVERED SOONER THAN IF THE DRAINAGE LINES WERE DIRECTLY CONNECTED TO THE SEWERS



strainer on the inlet end. In order to make the strainer of large size and capacity, a reducing elbow can be used to advantage, the strainer part being attached to the larger opening of the elbow. The strainer is shown at *H*.

The normal water level in the tank is shown at *W* and the oil level at *O*.

The sizes of the drainage pipes and dimensions of the tank or receiver depend on local conditions and the amount of drainage. The water overflow pipe should be of ample capacity in order that the water level will not rise and permit water to flow with the oil into the oil outlet pipe.

In case the pump or ejector is of large capacity and discharges a large amount of drainage intermittently, the arrangement of the tank can be modified by eliminating the baffle compartment and having a discharge tank or reservoir of large capacity above the water line in the separating tank. The discharge reservoir should have an outlet pipe of proper size to permit the drainage water to flow into the separating tank at a rate that can be handled by its water outlet.

Plant Engineers Find Photographs Useful in Recording Valve Locations

W. H. Wilson, Maintenance Engineer, Sees Helpful Ideas in Practice of Gas Company

The use of photographs to record valve locations is a very interesting subject to me, especially at this time. We have our troubles off and on, locating underground shutoff valves. In the past few weeks we have had more than the usual share of this work. Many of our underground valves are on water mains and it is not unusual to find a pile of lumber, rubbish or other material on the valve covers. Frequently when roadways are paved, the valve covers are covered over. Trouble in locating valves is increased when the ground is covered with snow. It may seem strange, but one can walk by a valve cover near a roadway every week for several years and know, apparently, just where it is, but try and find it with a shovel when the ground is covered with snow!

Indicator posts for valves cannot be used at all locations on account of being in the way of traffic. Any practical way of locating underground shutoff valves would be welcomed by industrial plant engineers.

A blue print showing the location of the underground pipe lines, with the valves, together with the locations of the buildings, is used at our plant. Many of the valves located on this print have dimension lines from the nearest buildings, or other fixed points. As our plant has been in operation for a number of years it is not unusual for alterations to be made, new buildings erected and other conditions necessitating the relocation of the valves. Sometimes these changes are noted on the print and sometimes they are not. This naturally adds to the pleasure of shutting down a section of the mains when a break occurs. Photographs of these valve locations would be of material aid. Photographs for this purpose should include additional information, which could be noted on the back:

Size of valve.
Name of manufacturer.
Manufacturer's figure number or class number, if any.
Kind of valve (screwed, flanged, bell ends, spigot ends).
Open to the left or open to the right.
Date installed.
Date last time valve stem packing box was packed.
Any particular or out of the ordinary supply that the valve controls. For example, we have underground valves that shut off only one hydrant, others that to close them means to shut off the cooling water to large air compressors, which would endanger an expensive machine.
Distance the valve is located from certain buildings, hydrants, curb, street intersection, telephone poles or other fixed objects.

Photographs Used to Check Parts of Reseating Machine

Some years ago we had photographs made of our valve reseating machine together with its numerous attachments, such as the cutters of the different sizes and other parts. This was a help to the tool room attendants in checking up the proper kind and number of parts. Prior to this the machine was often turned in with a cutter, spanner wrench or extension stem missing.

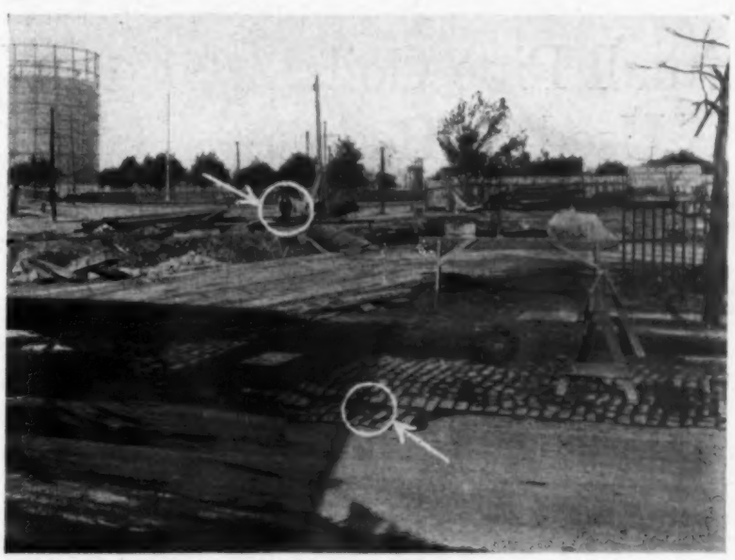
Avoiding Condensation in Laundries

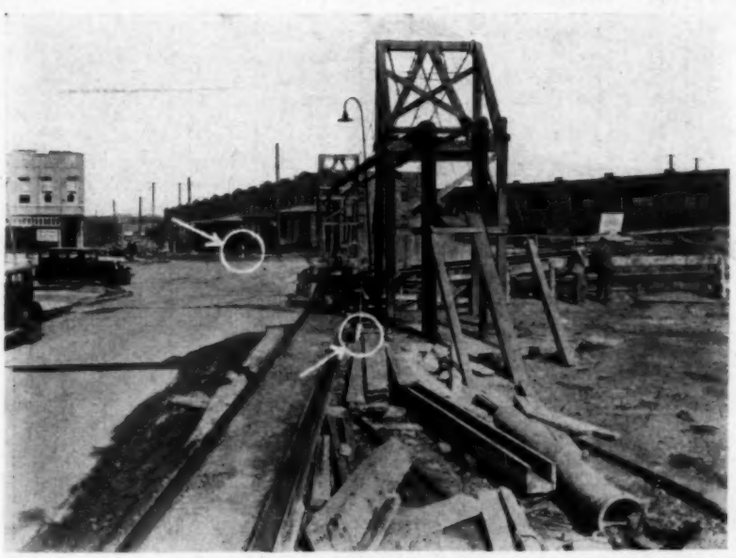
There are many places in a laundry where air conditioning can be used to advantage. Frequently, the requirements can be met satisfactorily with one or more unit conditioners.

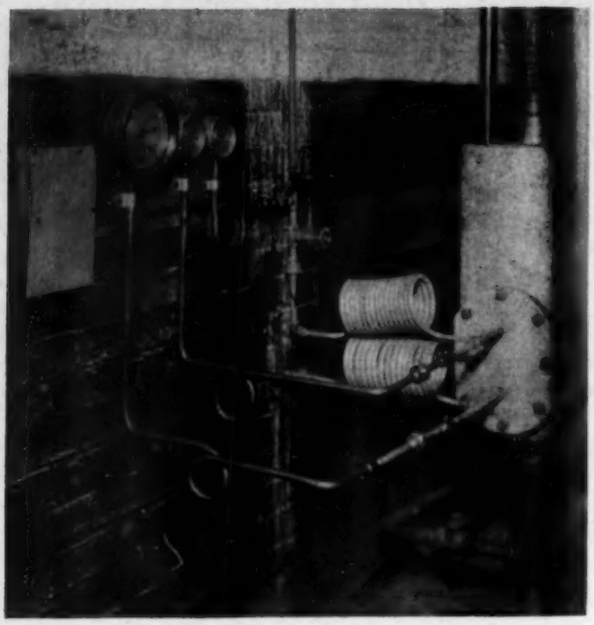
It is the writer's opinion that what a good many laundries need is a first class ventilating system; one that not only removes the steam and vapors nearest the source of their origin, but also supplies fresh air—not too cold, of course—through a good distributing system.

Condensation can be avoided in several ways, and usually we think of warming the surfaces against which the air comes in contact, although quite often we go about it the other way and lower the moisture content of the air. In any event, a good movement of air over a surface retards the point at which condensation begins. This is one of the contributing factors in obtaining satisfactory results with introduction and distribution of fresh, warm air.—*H. W.*

PHOTOGRAPHS SHOWING THE LOCATION OF VALVES IN THE MAINS OF THE BROOKLYN UNION GAS COMPANY. SUCH PHOTOGRAPHS FORM IMPORTANT RECORDS FOR INDUSTRIAL PLANT ENGINEERS ALSO







ONE OF THE ORIGINAL TEST SET-UPS

Small Pipe Coils Used to Reduce Steam for Heating

FOR some eight years the problem of maintenance of devices and valves to control steam for heating was a subject that caused considerable worry on the part of the writer. Let me present a problem, to be followed by the answer as finally worked out.

The heating load consists of:

Office building containing approximately 8,000 sq. ft. cast iron radiation, requiring a maximum of 1,920,000 Btu per hr. Two hot water storage tanks (150 gallons each) supplying water for domestic use.

Other factors affecting the problem are:

Temperature of office required: 70 F.

Temperature of water required: 160 F.

Medium of heat available: Steam at 140 lb. from central plant 7,500 ft. away and conveyed in insulated lines.

Character of steam: Very wet.

Condensation to be metered and wasted to sewer. Not economical to return to boiler plant because Btu value of condensate very low and cost of boiler make-up water nominal.

Type of system: Two pipe gravity return. Fractional inlet valves to radiators. Thermostatic return traps.

Air eliminated in service room through thermostatic float type eliminators.

Pressure required on system for zero weather: Two pounds on gage located in service room on the expansion header.

The idea was conceived that a reduction in pressure could be brought about by a predetermined manually operated control.

Engineering practice and mathematical formulas for flow of steam through given lengths of very small pipes were studied but they did not seem to accomplish what was wanted; after a long painful period of "cut" and "fit" and "try" a system of control such as shown in Fig. 1 was evolved and has served the purpose.

A careful analysis of the control unit will reveal a 2-in. high pressure header on the left and on the right a 5-in. low pressure expansion chamber or header.

At first it was thought that it would be necessary to care for a large amount of condensation at the bottom of the expansion header, and a valved outlet was put on for that purpose. However, it is interesting and surprising to note that the steam carries the entrained water up through the supply mains and from there it finds its way to the returns.

In between these headers will be noted seven friction coils made from a small size extra heavy steel pipe, fabricated while cold, by turning a section of straight pipe around a piece of 2½-in. pipe (much the same as you would make a spring.)

At the time this device was installed the steam traps that were previously used to care for the condensation were done away with and a change made on the returns such as is shown in Fig. 2.

Use of Water Leg

A simple water leg or seal, 10 ft. long (4 ft. in the ground and 6 ft. out) is introduced as a means of safety

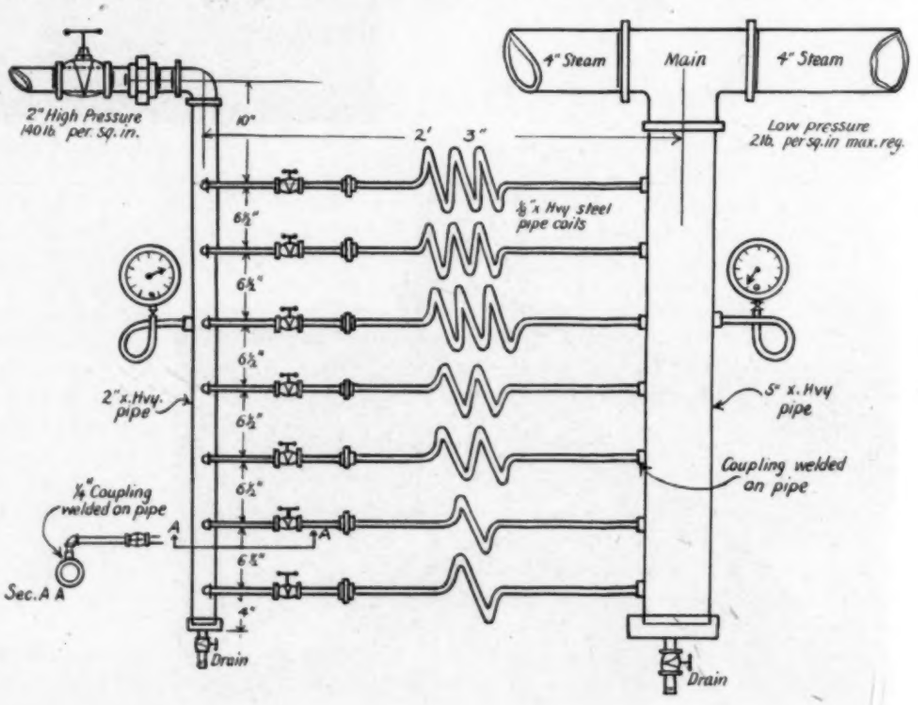


FIG. 1—FRICTION COIL PRESSURE CONTROL MANUALLY OPERATED FURNISHING STEAM FOR 8,000 SQ. FT. RADIATION; 1,920,000 BTU PER HR.

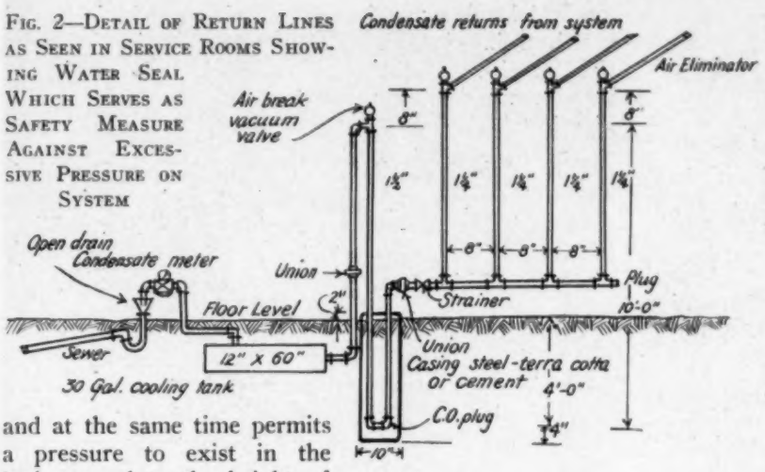


FIG. 2—DETAIL OF RETURN LINES AS SEEN IN SERVICE ROOMS SHOWING WATER SEAL WHICH SERVES AS SAFETY MEASURE AGAINST EXCESSIVE PRESSURE ON SYSTEM

and at the same time permits a pressure to exist in the mains equal to the height of the column of water. This is 4.34 pounds in this case. From this point the seal discharges into a 30-gal. cooling tank and from there overflows into an open funnel that is set into the sewer trap, as shown in Fig. 2.

In regard to water legs or water seals, it is well to install them in such a manner that they can be cleaned easily. A plug—as shown in Fig. 2—is often sufficient; however, a nipple and valve make it handier, providing, of course, that the end of the loop is accessible.

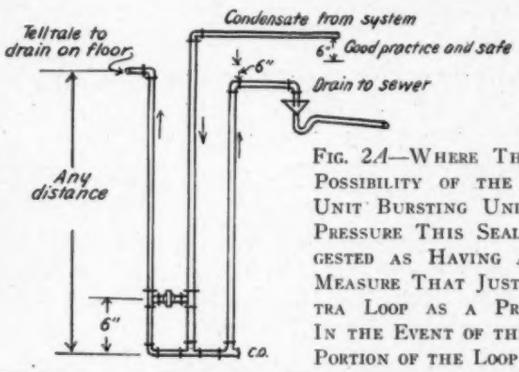


FIG. 2A—WHERE THERE IS A POSSIBILITY OF THE HEATING UNIT BURSTING UNDER HIGH PRESSURE THIS SEAL IS SUGGESTED AS HAVING A SAFETY MEASURE THAT JUSTIFIES EXTRA LOOP AS A PRECAUTION. IN THE EVENT OF THE BOTTOM PORTION OF THE LOOP STOPPING UP THEN THE CONDENSATE WOULD OVERFLOW AT TELL-TALE WHICH WOULD INDICATE NECESSITY OF REPAIRS

If it becomes necessary to extend one end below floor level it should be set in a tube or casing of some kind so that the loop may be removed without disturbing the floor. If the lines are known to be dirty it is good practice to put in a strainer as located in Fig. 2.

Operation of System

When heat is required in the building someone simply turns on one or more valves. In the event of too many being turned on it immediately becomes apparent by a bubbling over of the water and steam at the funnel or outlet to sewer. It does not bubble over very often and the extreme load that the system requires is easily taken care of by the seven coils. Shorter and fewer coils would accomplish the same thing, but the same degree of refinement of pressure would not be possible.

Method of Supplying Hot Water

Fig. 3 shows the application of this method to heating hot water. In the drawing can be seen one of two 150-gal. storage tanks heated with a small pipe. The heating coil serves a double purpose; producing a certain amount of friction and at the same time acting as the heating element. To further reduce the high pressure to atmospheric pressure a considerable longer coil is introduced as will be noted.

The problem of preventing the water from becoming too hot was solved by installing a thermostat in the tank, having it actuate a ¼-in. valve which was located on the outlet instead of the inlet. It was found advantageous to put it where the wire cutting action of the steam would not harm the valve seat or stem as it would if the valve were put on the inlet. It is interesting to note that the heating coil shown in Fig. 3 is only 12 ft. long and still it heats water at the rate of a little over three gallons per minute. With the tank full of cold water and the supply valve closed against drawing, the entire tank is heated in approximately 50 minutes.

Several other practical applications have been made of this idea. For instance, in a small isolated office heat was required for three radiators (210 ft. cast iron radiation) and this method was applied and is giving satisfaction. Application of the principle has also been made to a pipe coil used for heating a lumber drying shed; in this case there is no water leg. (High pressure steam at one end and condensation to the sewer at the other.)

On another installation consisting of a large service or work room a small unit heater was placed in one corner 10 feet from the floor and the steam broken down in pressure by the aid of friction coils, and the condensation is passed through a 30-gal. storage tank used to furnish hot water for a couple of lavatories. The heater is by-passed so that hot water can be had in the summer when the office heating system is not running.—*T. D.*

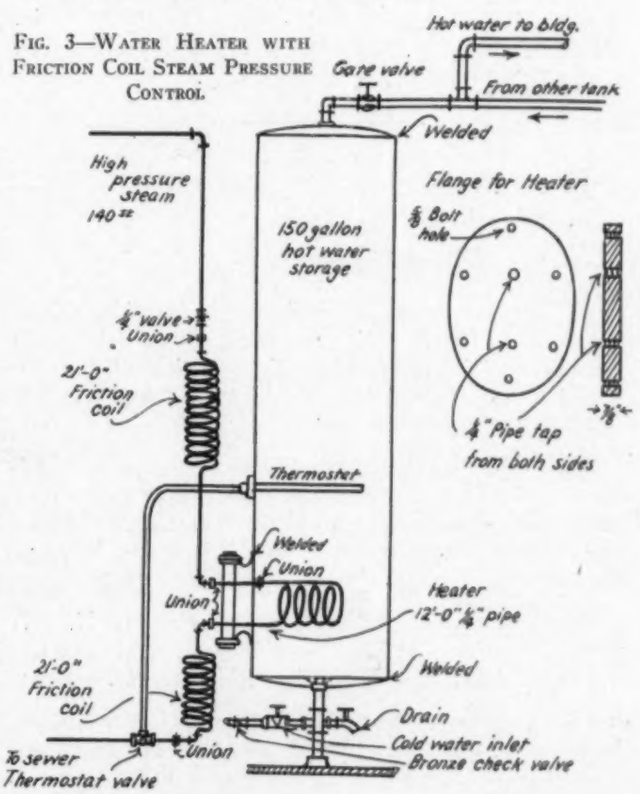


FIG. 3—WATER HEATER WITH FRICTION COIL STEAM PRESSURE CONTROL

Conventions and Expositions

International Association for Testing Materials: First International congress, Sept. 6-12; The Swiss Federal Polytechnicum, Zurich, Switzerland. General secretary, Prof. M. Ros, 27, Leonhardstrasse, Zurich.

National Association of Power Engineers: Annual convention and exposition, September 7-11; Convention Hall. Kansas City, Mo. Secretary, F. W. Raven, 417 S. Dearborn st., Chicago.

American Welding Society: Fall meeting, Sept. 21-25; Copley-Plaza Hotel, Boston, Mass. Secretary, M. M. Kelly, 33 W. 39th st., New York City.

American Gas Association: Annual convention and exhibition, Oct. 12-13; Municipal Auditorium, Atlantic City, N. J.

Southern Power and Machinery Show: October 19-24; Textile Hall, Greenville, South Carolina.

Third International Conference on Bituminous Coal: November 16-21; Carnegie Institute of Technology, Pittsburgh, Pa. Chairman, Thomas S. Baker, President, Carnegie Institute of Technology.

American Society of Mechanical Engineers: Annual meeting, Nov. 30-Dec. 4; New York City. Secretary, Calvin W. Rice, 33 W. 39th st., New York City.

American Society of Heating and Ventilating Engineers: Annual meeting, Jan. 25-28; Cleveland, Ohio. (Joint meeting with A. S. R. E.) Secretary, A. V. Hutchinson, 51 Madison ave., New York City.

International Heating and Ventilating Exposition: Jan. 25-29; Cleveland Auditorium, Cleveland, Ohio. Manager, Charles F. Roth, International Exposition Co., Grand Central Palace, New York City.

Recent Trade Literature

Air Heaters: Nichols Products Corporation, 12953 Greeley ave., Detroit, Mich.; four-page pamphlet featuring units for air heating adapted for use in battery; suggested uses include industrial plants covering areas of from 10,000 to 500,000 square feet.

Fans: The American Propeller Company, Inc., South Bend, Ind.; four-page bulletin describing propeller type fans for man cooling; two-blade fans have capacities from 900 to 82,000 c.f.m.; four-blade, from 3,200 to 47,500.

Filters: Staynew Filter Corp., Rochester, N. Y.; folder featuring portable, dry, air filter for window use in offices, club rooms, hospitals, residences, etc.

Pipe Couplings: Victualic Company of America, 26 Broadway, New York City; 32-page loose-leaf data book on flexible pipe couplings for steel, wrought iron, and cast iron lines in diameters from ¾ to 60 in. Information on pipe and fittings is included, and uses in the petroleum and mining industries are discussed. Marine pipe installations are also considered.

Pipe Fittings: Taylor Forge & Pipe Works, P. O. Box 485, Chicago, Ill.; 32-page data book giving engineering information on seamless steel pipe fittings for welding, including elbows, tees, reducing tees, bull plugs, reducing nipples, and high-hub forged steel welding flanges.

Steam Generators: General Heat and Power Corp., 730 Fifth ave., New York City; 10-page loose-leaf folder describing boilers and burners using oil or gas; no stack is required. Heating units are available for capacities from 750 to 20,000 sq. ft. of radiation and power units from 25 to 300 h.p. Drawings and dimensions are included.

Steam Purifiers: Centrifix Corporation, 3029 Prospect ave., Cleveland, Ohio; four-page folder describing features of a device for separating liquids and solids from moving steam, vapor, gas and air. Uses as an internal steam purifier, line purifier, exhaust line purifier, compressed air cleaner, receiver purifier, suction intake purifier, dust separator for gas lines, and evaporator purifier are discussed.

Stokers: Simplex Oil Heating Corporation, 30 Church st., New York City; four-page bulletin describing hydraulically operated underfeed stokers for all types of boilers from 50 to 500 h.p. The stokers are of the single retort, side dump type.

Tanks: H. O. Swoboda, Inc., 3400 Forbes st., Oakland Sta., Pittsburgh, Pa.; eight-page bulletin on electrically heated melting and coating tanks suitable for asphalts, tars, compounds, varnishes, oils, etc.

Unit Air Washers: Air-control Systems, Inc., 2240 N. Racine ave., Chicago, Ill.; eight-page pamphlet giving complete information about unit air washers suitable for humidifying up to 40,000 cu. ft. of space in offices, residences, etc. As cooling or heating units, they are ample for one room. Methods of installation, specifications, pointers on operation, etc., are also covered.

Unit Heaters: American Blower Corporation, Detroit, Mich.; 44-page application guide containing complete information on planning and installing unit heater jobs and showing numerous typical lay-outs. Topics such as boiler selection, motors, temperature control, etc., are also considered.

Valves: The Ludlow Valve Mfg. Co., Troy, N. Y.; six-page bulletin describing features of a valve giving multi-stage reduction. Uses as a blow-off valve, throttle valve and with steam, air, gas, water, oil and viscous liquid pipe lines are suggested. Dimensions and price list included.

Welders: Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.; data sheet devoted to 100 ampere welder with short wave arc control for use in erecting duct work, etc. Operation, dimensions, application, performance, etc., are discussed.